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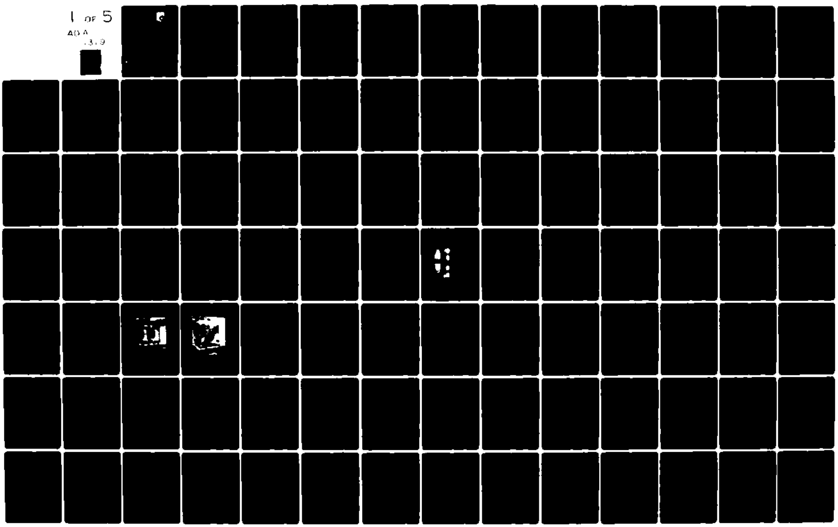

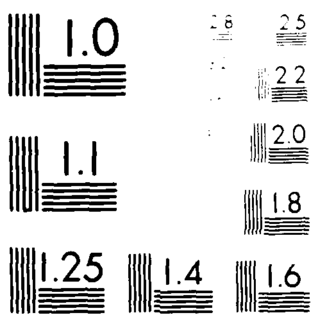
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FIREPROOF BRAKE HYDRAULIC SYSTEM

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TECHNICAL REPORT AFWAL-TR-81-2080
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The feasibility of a fireproof two-fluid brake hydraulic system which utilizes nonflammable chlorotrifluoroethylene (CTFE) fluid in a dead ended brake hydraulic system mechanically isolated from the primary MIL-H-5606 hydraulic system was studied. Experimental brake system hardware was designed, built and laboratory tested. Results indicate that the fireproof two-fluid brake hydraulic system concept is feasible, and the basic operation, control characteristics and stability of the brake system are not affected by the two-fluid

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20. ABSTRACT (continued)

configuration. The increased density of the CTFE fluid does cause the hydraulic system to respond slower resulting in longer aircraft stopping distances. However, analysis indicates that the performance lost by changing to the CTFE fluid can be regained by increasing brake hydraulic line sizes and retuning the antiskid control box.

FOREWORD

This report was prepared by S. M. Warren, and J. R. Kilner of the Boeing Military Airplane Company under USAF Contract F33615-80-C-2026. This contract was accomplished under Project Number 31453032, Fireproof Brake Hydraulic System. The work was conducted under the direction of the Power Systems Branch, Aero Propulsion Laboratory, Air Force Wright Aeronautical Laboratories, Wright Patterson Air Force Base, Ohio. Mr. W. B. Campbell (AFWAL/POOS) was Project Engineer. The objective of this contract was to investigate the feasibility of a fireproof two-fluid brake hydraulic system that will reduce potential aircraft fires in the wheel well and landing gear area. This report describes all essential aspects of the work performed in completing this contract. The work was performed from June 1980 to June 1981.

The authors are indebted to E. T. Raymond of the Boeing Company for his technical assistance throughout the design and testing of the two-fluid brake system.

The authors are also indebted to Mr. W. B. Campbell (AFWAL/POOS) and Mr. C. E. Snyder (AFWAL/MLBT) for their contributions throughout the program.

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SECTION I

INTRODUCTION

The Air Force has become increasingly concerned about the danger and dollar loss caused by aircraft hydraulic fires. A major cause of these fires is the ignition of hydraulic fluid on hot surfaces. During the 1970 to 1975 time period 63 percent of the hydraulic fluid fires occurred in the wheel well and/or landing gear area. Most of these fires were related to the ignition of hydraulic fluid on hot brakes.

The hydraulic fluid currently used on most military aircraft is a petroleum-based mineral fluid per MIL-H-5606 which has a low manifold ignition temperature and high heat of combustion, and burns quite readily. Although this fluid is used throughout the aircraft, the brake, steering and landing gear hydraulic actuation systems are statistically the most vunerable. For example, when a hydraulic failure occurs in which the petroleum-base hydraulic fluid contacts a hot brake, rapid ignition of the fluid occurs creating intense heat which ignites other fuel sources (such as the tire) that sustain the fire after the hydraulic fluid source is depleted.

In an effort to reduce the occurrence of aircraft hydraulic fires the Air Force initiated a program to develop a nonflammable hydraulic fluid. This effort led to the development of chlorotrifluoroethylene (CTFE) base hydraulic fluids. Although virtually nonflammable, the principal disadvantage of the CTFE hydraulic fluid is its high specific gravity (density) which is 2.11 times that of the MIL-H-5606. In addition, the fluid is not compatible with Buna N seals which is the elastomer material commonly used in MIL-H-5606 hydraulic fluid systems, and CTFE fluid cost for high production quantities is very high compared to aircraft hydraulic fluids presently in use.

The use of CTFE fluid in aircraft hydraulic systems would greatly alleviate the fire danger and result in a significant improvement in aircraft safety. However, replacing MIL-H-5606 with CTFE throughout the entire aircraft hydraulic system would result in a significant weight penalty due to the

increase in fluid density (e.g., +1700 lbs for the YC-14). This weight penalty can be reduced to approximately 64 lbs for a cargo/transport aircraft and 30 lbs for a fighter aircraft by employing a two-fluid hydraulic system concept in which the CTFE fluid is used only in the immediate area of the landing gear and brakes.

The Fireproof Brake Hydraulic System program was initiated by the Air Force to develop and evaluate such a two-fluid brake hydraulic system. The KC-135 was selected as the study aircraft representing a typical modern cargo aircraft. A laboratory mockup of the KC-135 brake hydraulic system was converted to a two-fluid configuration and laboratory tests were conducted to determine the system's operational characteristics and the resulting impact on aircraft braking performance. The CTFE fluid used in the study was Halocarbon AO-2.

The Fireproof Brake Hydraulic System program was divided into the six tasks shown in Table 1. The two major tasks in the program were the design of the two-fluid brake hydraulic system (Task 2) and the testing to determine the impact of the two-fluid concept on brake system performance (Task 5).

The initial effort of the program was to configure and analyze a two-fluid brake hydraulic system for the KC-135. Special attention was given to designing a system which:

(1) provides positive separation of the two hydraulic fluids (MIL-H-5606 from the CTFE fluid),

(2) is safe, reliable and easily maintained, and

(3) minimizes changes to the existing brake hydraulic system thereby reducing system costs and increasing the feasibility of retrofitting the two-fluid system on existing aircraft.

After configuring the two-fluid brake hydraulic system, a laboratory mockup of the KC-135 two fluid brake system was constructed. Hydraulic components were modified or fabricated as necessary for testing of the two-fluid system. The laboratory two-fluid brake hydraulic system was combined with an antiskid

Table 1 FIREPROOF BRAKE HYDRAULIC SYSTEM PROGRAM TASKS

Task 1 Acquisition of Brake Components.

A modern-cargo type aircraft brake system was selected for conversion to a two-fluid system. The brake system components necessary for laboratory testing were acquired.

Task 2 Component Assessment and Redesign for the Two-Fluid System.

The components and system modifications necessary to convert the brake system to the two-fluid configuration were determined. An analytical assessment of the hydraulic system was performed to determine system level design changes.

Task 3 Component and System Test Plan

A test plan, designed to define the performance of individual components and the entire brake system, was developed. The test plan was submitted to the Air Force for approval.

Task 4 Component Modification and Test

Brake system components were modified to function in the two-fluid system. Tests were conducted to assure proper function of each modified component.

Task 5 Component Installation and System Test

The modified components were installed in the two-fluid brake hydraulic system. Tests were conducted to determine the system operation and stopping performance of the two-fluid brake system.

Task 6 Reliability/Maintainability Study

The impact of the two-fluid configuration on brake system reliability and maintainability was estimated.

brake control unit and a hybrid computer airplane simulation of the KC-135 to determine the impact which the two-fluid configuration has on brake system operation and aircraft stopping performance.

This report presents the results of the 12 month effort conducted to determine the feasibility of the two-fluid fireproof brake hydraulic system concept. The report contains 5 sections. Section II decribes the two-fluid brake hydraulic system design and the required component and system modifications to a conventional system. In Section III, the results of the component and system performance laboratory tests are described. The two-fluid brake system performance is compared with the performance of the standard single fluid MIL-H-5606 (baseline) brake system.

The reliability (failure rate) and maintainability (servicing rates) of the two-fluid brake system are discussed and compared to the baseline brake system in Section IV.

Conclusions and recommendations based upon the design, analysis and testing reported in Sections II, III and IV are presented in Section V.

Several appendices are included to present the Interim Technical Report and the test plans, raw component and system laboratory test data, a description of the airplane simulation used during the system tests, and other related information.

SECTION II

TWO-FLUID BRAKE HYDRAULIC SYSTEM DESIGN

2.1 PROGRAM OBJECTIVE

The objective of the Fireproof Brake Hydraulic System study was to determine the feasibility of a two-fluid brake hydraulic system which uses a nonflammable CTFE base hydraulic fluid in the immediate area of the landing gear and brakes.

2.2 PROGRAM APPROACH

The two-fluid brake system was evaluated by comparing its system dynamic performance and resultant aircraft stopping distances with the performance and stopping distances of the standard single-fluid MIL-H-5606 brake system (termed the baseline brake system).

To accomplish this end the baseline brake hydraulic system was redesigned to function as a two-fluid system. Hardware was fabricated and existing components modified to convert the baseline system to the two-fluid configuration. The two-fluid brake hydraulic system was assembled and laboratory tests performed to define the operational characteristics and performance of the modified system. The same sequence of tests were performed for the baseline system for a comparison and evaluation of the two-fluid brake hydraulic system.

2.3 STUDY BRAKE HYDRAULIC SYSTEM

The KC-135 was selected as the study aircraft. The KC-135 brake system is representative, and contains all the features, of a modern cargo aircraft brake control system. A schematic of the KC-135 brake system is shown in Figure 1. The aircraft has two four wheel truck type main landing gear with paired wheel brake control. That is, the brake pressure associated with each forward and aft wheel pair on one side of the truck is controlled by a single antiskid valve and antiskid control system. The portion of the brake system associated with a single-tandem-wheel pair is shown in Figure 2.

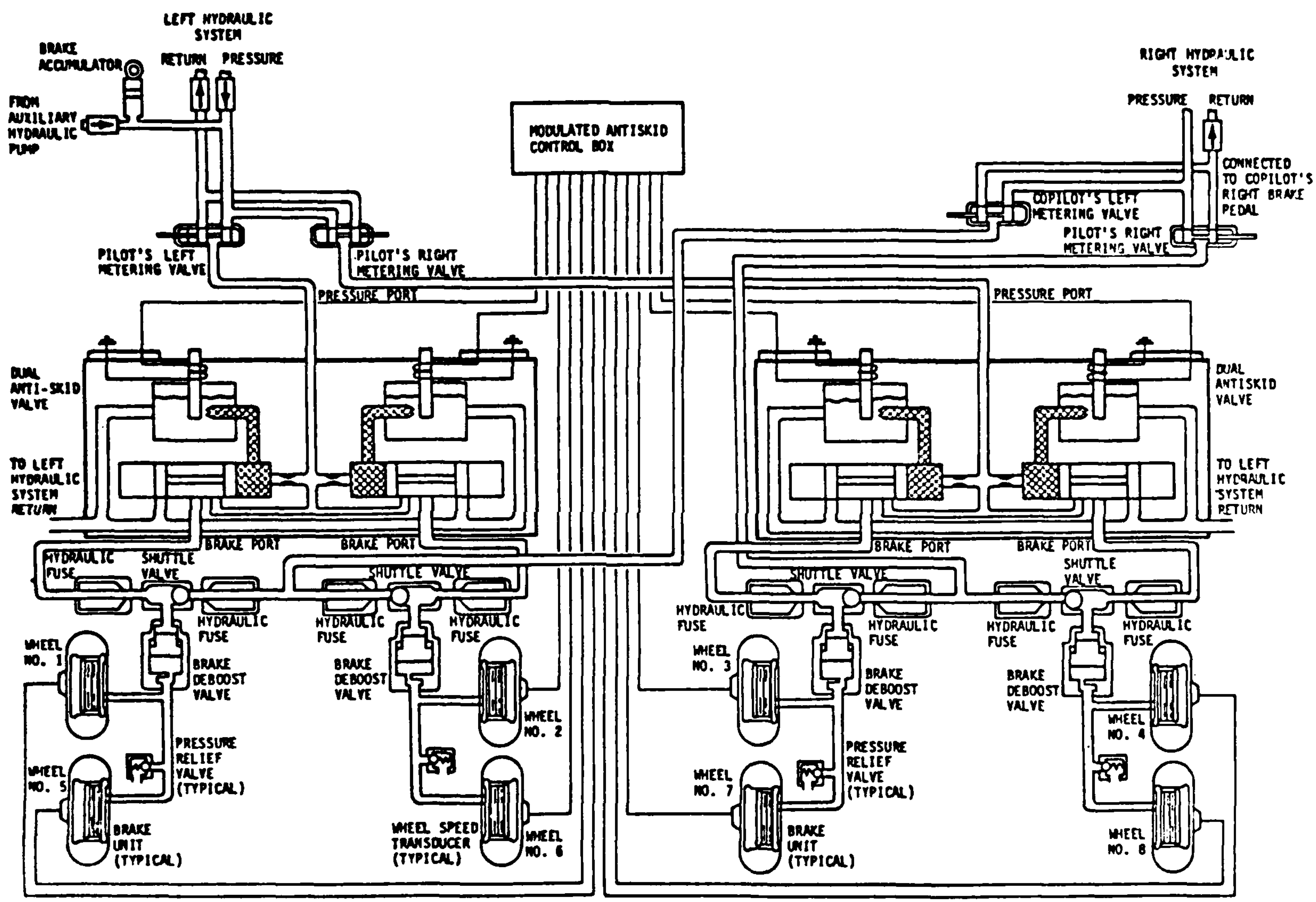


Figure 1 KC-135 Brake System Schematic

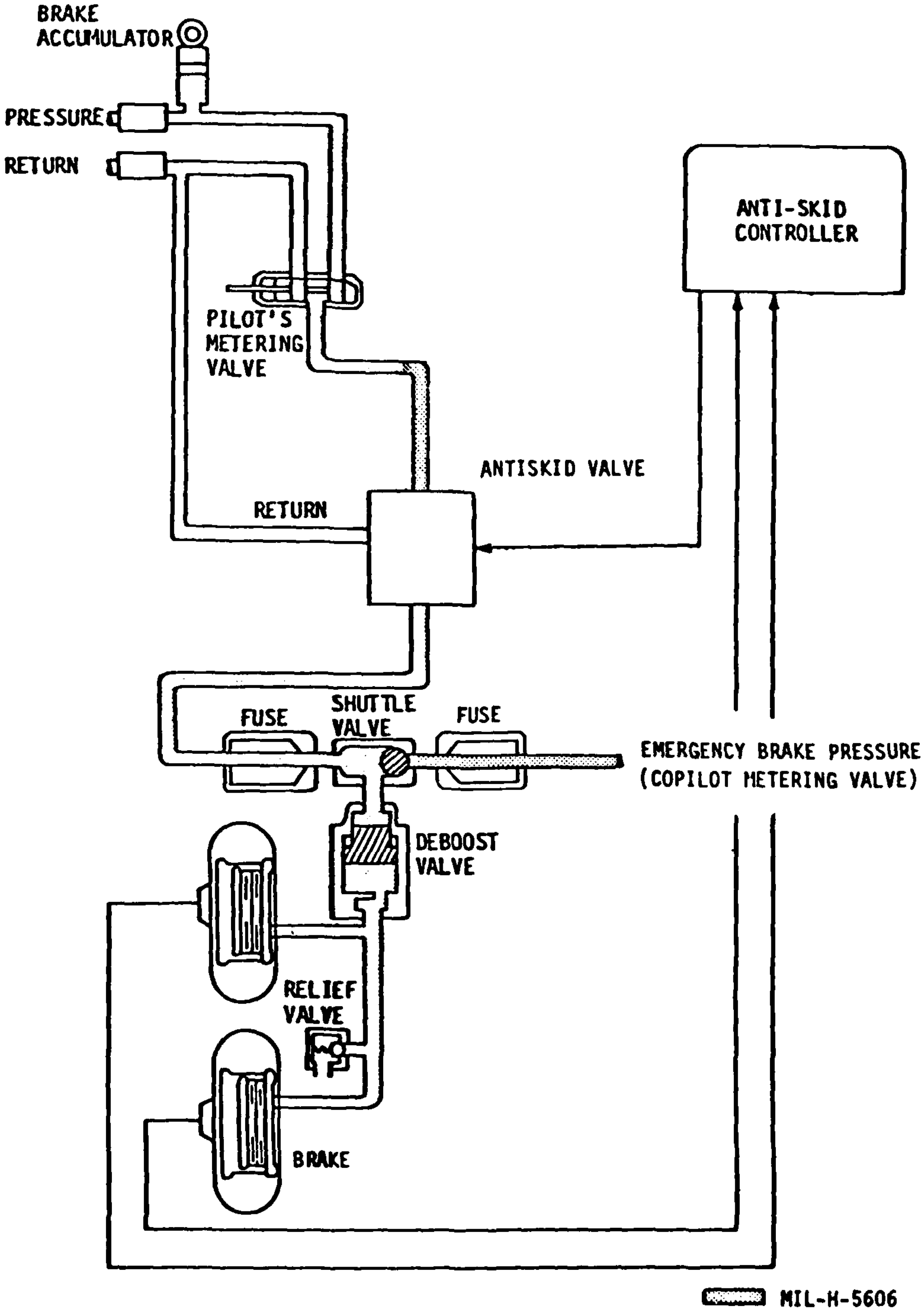


Figure 2 KC-135 Brake System, Single-Tandem-Wheel Pair

2.4 TWO-FLUID BRAKE HYDRAULIC SYSTEM DESIGN OBJECTIVE

The overall objective of the system design effort was to define a two-fluid brake hydraulic system configuration which was then used to determine and evaluate the effects that the two-fluid concept has on the brake system operation and aircraft performance. Specifically, the task involved redesigning the existing KC-135 brake hydraulic system to incorporate the two-fluid concept.

The system design effort and the selection of a two-fluid system configuration was approached with the following goals in mind:

(1) To minimize the changes to the existing brake hydraulic system,

(2) To keep system fabrication and material costs low, and

(3) To design system changes that can be easily retrofitted on existing aircraft.

2.5 TWO-FLUID BRAKE HYDRAULIC SYSTEM DESIGN REQUIREMENTS

The two-fluid brake hydraulic system was designed to meet the following component and system requirements. The design shall:

(1) Provide positive and reliable separation of the two hydraulic fluids (MIL-H-5606 from the CTFE fluid).

(2) Operate in a temperature range of -65 degrees F. to 160 degrees F. (the range of brake system operation).

(3) Provide a CTFE fluid replenishment system which accounts for volumetric changes due to temperature, brake wear and normal fluid loss (such as seal leakage).

(4) Meet present aircraft reliability and safety standards.

2.6 TWO-FLUID BRAKE HYDRAULIC SYSTEM DESIGN

The two-fluid brake hydraulic system design effort was performed in three phases. First, several preliminary KC-135 two-fluid brake hydraulic system configurations were designed. These were evaluated based on the maintainability, reliability, cost and risk associated with each design. The best preliminary design was selected for detailed development.

During the second phase, a detailed design of the two-fluid brake hydraulic system was performed. Engineering drawings of each new or modified KC-135 component in the two-fluid brake hydraulic system (required for laboratory testing) were made. In addition, servicing procedures for the brake hydraulic system were formulated.

An analytical assessment of the KC-135 two-fluid brake hydraulic system was performed during the third phase to determine: (1) the effect which the proposed two-fluid system has upon the dynamic response of the brake hydraulic system, and (2) the modifications to the two-fluid system configurations which are required to achieve a dynamic response characteristic comparable to the standard KC-135 brake hydraulic system. The results of the analytical study are discussed in Section 2.7 while the final KC-135 two-fluid brake hydraulic system configuration is discussed below.

2.6.1 KC-135 TWO-FLUID BRAKE HYDRAULIC SYSTEM CONFIGURATION

The basic features of the KC-135 two-fluid system design selected for this study are discussed in the following paragraphs. A detailed explanation of the configuration may be found in Appendix A which is a reprint of the Interim Technical Report.

The KC-135 two-fluid brake hydraulic system design features a modified deboost valve as the fluid isolator separating the MIL-H-5606 and CTFE fluids. The deboost valve, shown schematically in Figure 3, is a pressure reducing device with a floating differential area piston, a flow through fluid replenishment valve and a replenishment pin. Normally the deboost valve functions as a pressure reducer, where the ratio of the output pressure to the input pressure

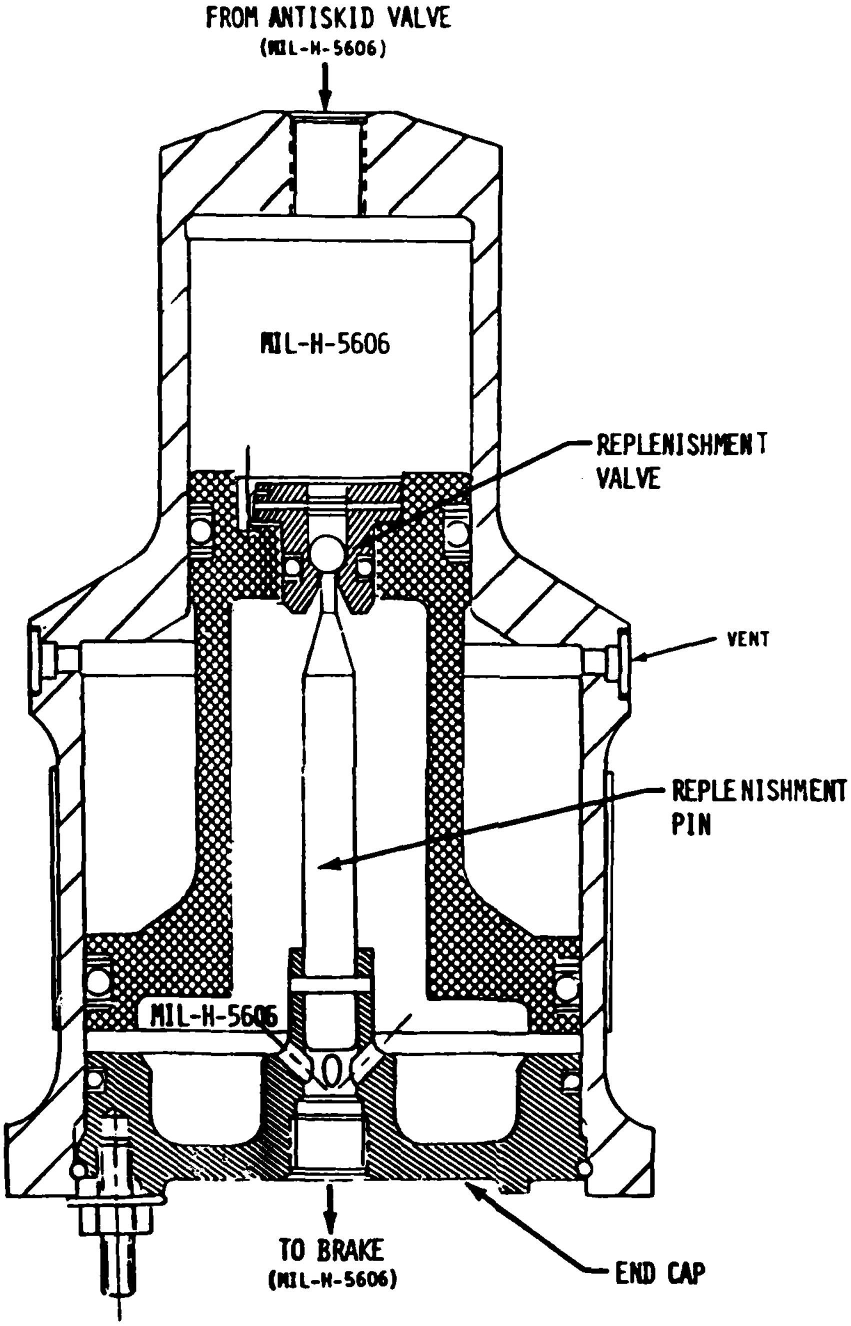


Figure 3 Deboost Valve

is 0.3217 under static conditions (e.g., 3000 psi pressure at the input is reduced to 965 psi at the output of the deboost valve). The replenishment pin and replenishment valve permit fluid to flow from high pressure to low pressure when makeup fluid is required in the normally isolated low pressure volume. Fluid isolation is achieved by removing the replenishment pin and replenishment valve and plugging the hole in the floating piston as shown in Figure 4.

The KC-135 two-fluid brake hydraulic system associated with a single tandem-wheel pair is shown in Figure 5. The principal features of the two-fluid configuration are:

(1) The isolation of the MIL-H-5606 fluid from the CTFE fluid with a floating piston device,

(2) A CTFE replenishment system to replace lost fluid, and

(3) The use of PNF O-ring seals in areas exposed to the CTFE fluid.

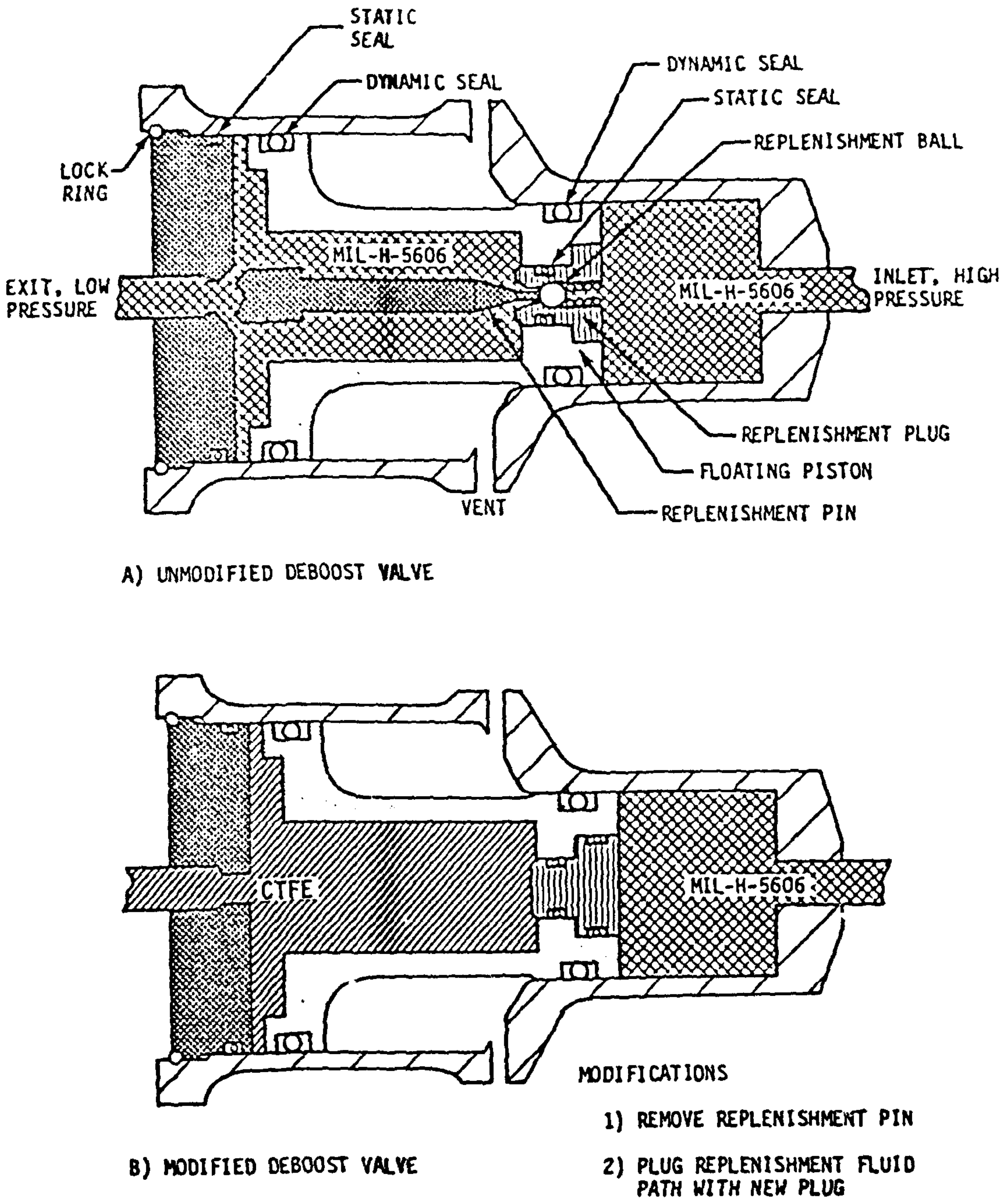
The specific modifications necessary to convert the KC-135 brake hydraulic system to the two-fluid configuration are:

(1) The elimination of the original flow path through the deboost valve,

(2) The addition of a separate CTFE fluid replenishment system, and

(3) The use of PNF O-ring seals in the CTFE portion of the hydraulic system.

The modifications to the deboost valve and the added CTFE replenishment system are shown in Figure 6. The original replenishment valve and pin in the standard KC-135 deboost valve (see Figure 3) have been removed and the hole in the piston plugged. These modifications eliminate the fluid flow path thru the piston converting the original deboost valve into a fluid isolator. The original end cap has been removed and replaced with a new end cap containing a bleed/output standpipe and a new replenishment valve. The standpipe is provided primarily for removing air from the deboost valve during servicing.



MODIFICATIONS

1) REMOVE REPLENISHMENT PIN

2) PLUG REPLENISHMENT FLUID PATH WITH NEW PLUG

3) INSTALL NEW SEALS

Figure 4 Fluid Isolator

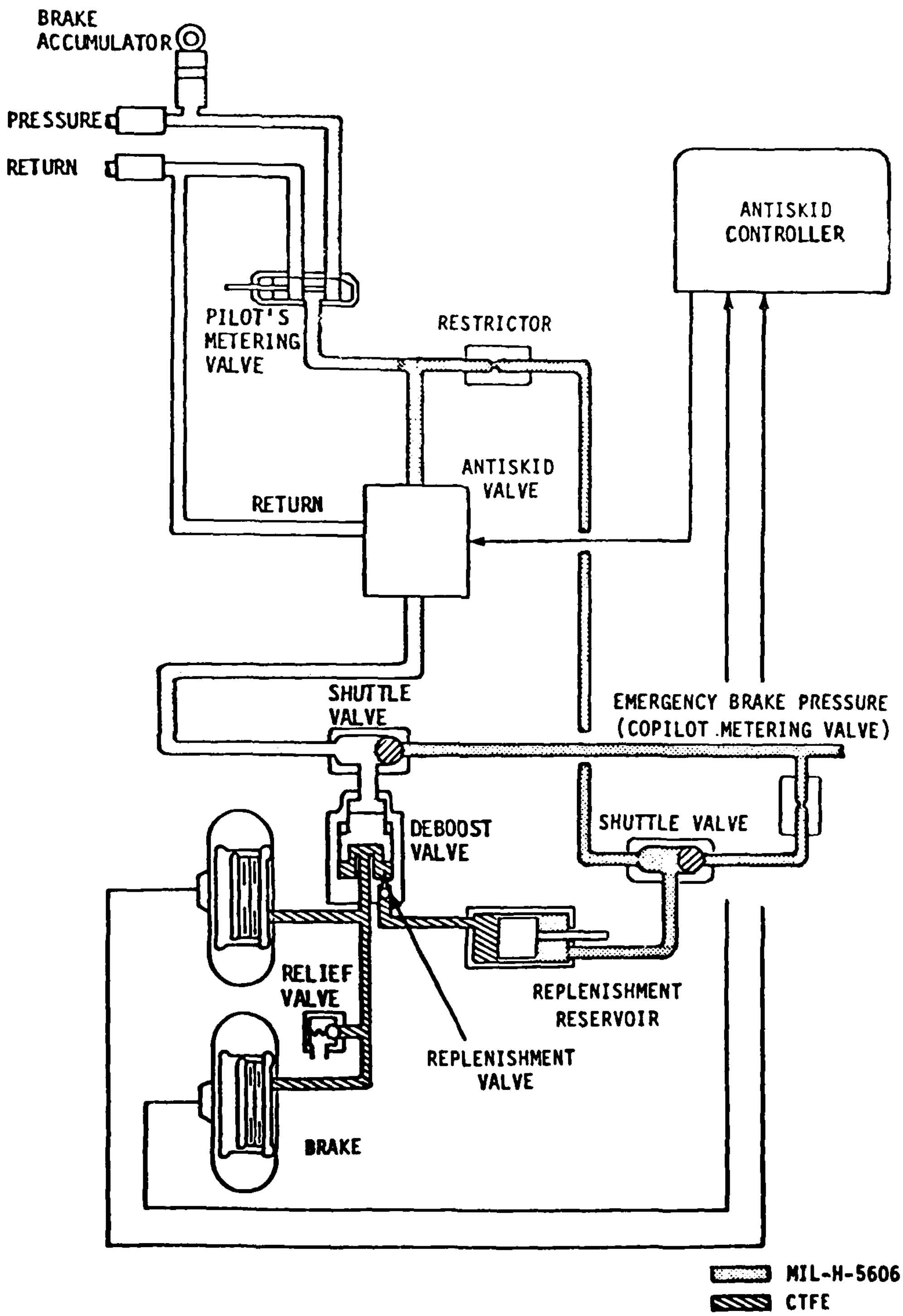


Figure 5 KC-135 Two-Fluid Brake System, Single-Tandem-Wheel Pair

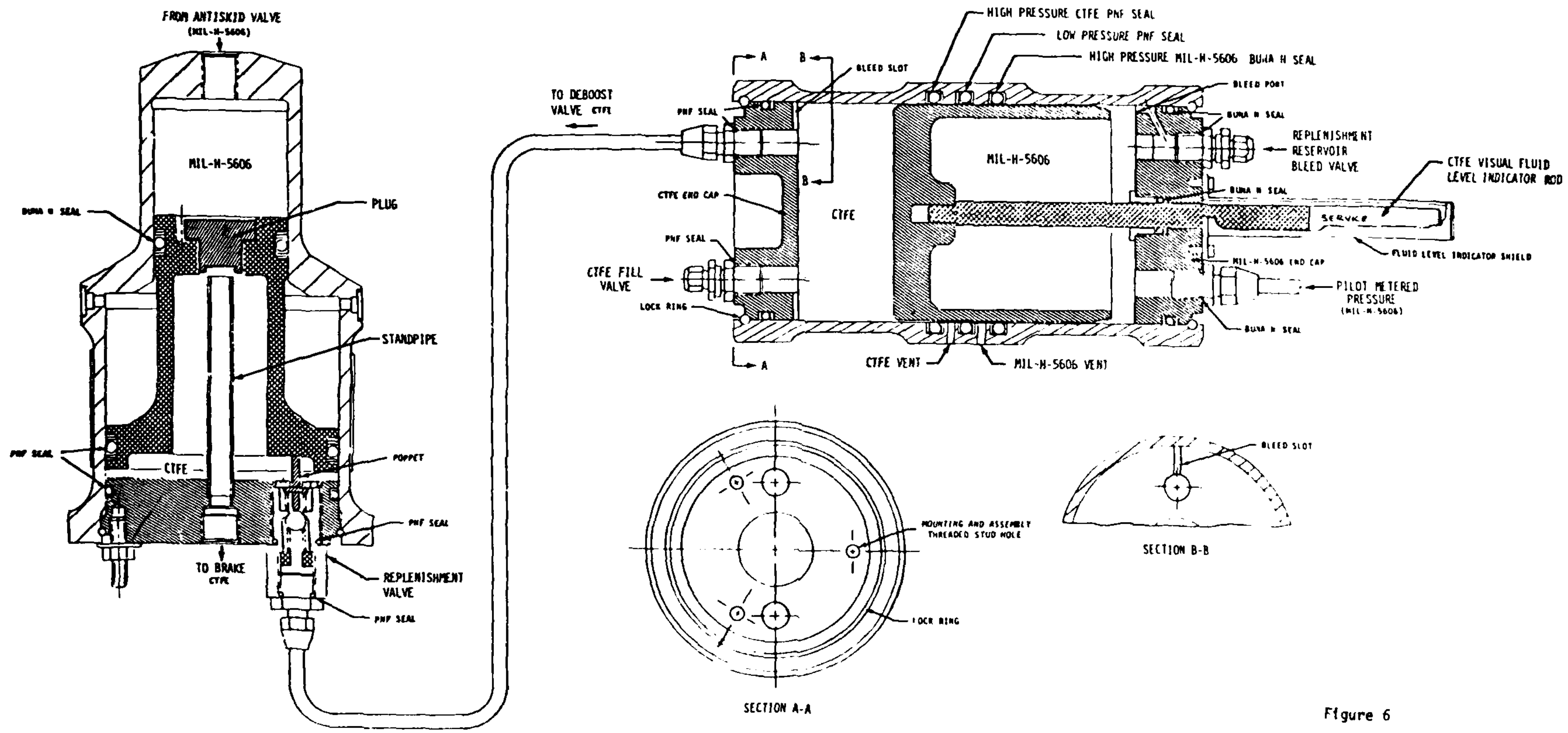


Figure 6

Modified KC-135 Deboost Valve

and CTFE Replenishment System

The CTFE replenishment system includes the replenishment valve and a pressurized fluid reservoir. The replenishment valve is a poppet actuated spring loaded ball type check valve which screws into the end cap. The replenishment reservoir contains a CTFE fluid volume which is pressurized by pilot metered brake pressure (MIL-H-5606 fluid). In the event of a hydraulic failure upstream of the shuttle valve the reservoir is pressurized by copilot metered pressure.

The floating piston two-fluid isolation concept is used in the reservoir design to separate the CTFE and MIL-H-5606 fluids. A fluid level rod is attached to the floating piston to indicate when CTFE fluid must be added to the replenishment reservoir. The volume of CTFE fluid (30 cubic inches per KC-135 two-wheel set) contained in the reservoir is sufficient to account for volumetric changes due to temperature, brake wear and normal fluid loss (such as seal leakage).

The KC-135 brake hydraulic system contains hydraulic fuses to protect against excessive fluid loss in case of a line burst. However, this protection is provided by the modified deboost valve and independent CTFE fluid replenishment reservoirs in the two-fluid system.

In addition to the modifications described above, the original nitrile seals used in the deboost valve and brake are replaced with PNF type seals. PNF seals must be used in areas exposed to CTFE fluid. PNF seals are also used in the replenishment reservoir as necessary.

It should be noted that the design described here is an example of a workable two-fluid system and does not necessarily represent the optimum design for the KC-135 aircraft. However, the design is sufficient and appropriate for determining the feasibility of the two-fluid brake hydraulic system concept.

2.6.2 KC-135 TWO-FLUID BRAKE HYDRAULIC SYSTEM OPERATION

The KC-135 two-fluid brake hydraulic system design discussed above was selected for the feasibility study because it has two distinct advantages. First, the brake hydraulic system is virtually unchanged; and second, it

functions exactly the same as the original system. No modifications have been made which affect or change the basic operation of the deboost valve or brake system.

During normal brake system operation, the original KC-135 brake system and the two-fluid brake hydraulic system are identical. The differences which exist between the configurations involve only the replenishment system. The original replenishment valve has simply been moved from the deboost valve piston to the end cap. Since the replenishment valve is closed in both systems during normal braking activity, the configurations are identical. Similarly, other brake system operating modes such as parking, refused takeoff, manual braking and emergency braking are not affected by the hardware modifications or configuration.

The operation of the two-fluid replenishment system is nearly identical to the original KC-135 system. The major difference is that the replacement fluid comes from different sources. In both systems, replenishment occurs when the deboost valve piston floats down toward the end cap (caused by a reduction of the fluid volume in the low pressure portion of the deboost valve) opening the replenishment valve. High pressure fluid enters the deboost valve replacing lost fluid. As the fluid volume increases the piston floats up closing the replenishment valve. The CTFE replenishment reservoir is pressurized by pilot metered brake pressure. The reservoir is only pressurized when braking is commanded.

2.6.3 SYSTEM SAFETY FEATURES

Several features have been included in the two-fluid brake hydraulic system configuration to improve system safety.

Each KC-135 tandem-wheel-pair (i.e., each independently controlled brake hydraulic subsystem) has an independent fluid replenishment system to prevent the total loss of aircraft braking capability in the event of a failure in the isolated CTFE portion of the two-fluid brake hydraulic system. For example, if a hydraulic line between the fluid isolator (KC-135 deboost valve) and brake bursts, only the braking capability and fluid associated with that wheel

pair is lost. Normal braking capability and replenishment capacity is maintained on the three other paired wheel sets.

The replenishment reservoir is pressurized normally by pilot metered pressure. In the event of a hydraulic failure upstream of the shuttle valve the reservoir is pressurized by the copilot metered pressured.

The use of pilot metered pressure to power the replenishment reservoir provides an additional safety feature. For example, if the MIL-H-5606 side of the replenishment reservoir was connected directly upstream of the pilot metering valve to the 3000 psi supply or if a high pressure air charged accumulator was used to power the reservoir, the brakes could be permanently pressurized and locked due to a leakage failure through the replenishment valve (see Figure 5). This type of failure is avoided since the reservoir pressure is dumped to return when the pilot releases the brakes.

2.6.4 SERVICING THE TWO-FLUID BRAKE HYDRAULIC SYSTEM

The KC-135 two-fluid brake hydraulic system has been designed for easy maintenance and servicing. The replenishment reservoir design includes MIL-H-5606 fluid supply and bleed ports, and CTFE fluid fill and output ports which aid servicing.

Filling and bleeding the brake system is accomplished by ground servicing of the MIL-H-5606 portion of the system followed by the CTFE portion of the system. Servicing the MIL-H-5606 portion is performed by applying maximum brake pressure and cracking the reservoir bleed valve to circulate MIL-H-5606 through the brake system. The CTFE portion is then serviced by opening the brake bleed valve and pumping CTFE through the reservoir and brakes. The deboost valve standpipe has been included to assure that all air is purged from the deboost valve. When the CTFE portion of the system is serviced the deboost valve piston is bottomed against the end cap, the replenishment valve is open and the standpipe is in the cavity of the plug (see Figure 6). As CTFE fluid is pumped through the brake system, air collects in the plug cavity. This air is forced down the standpipe and out the brake as the deboost valve volume fills with fluid.

Detailed fill and bleed procedures for both the MIL-H-5606 and CTFE portions of the two-fluid brake hydrualic system are given in Appendix A.

2.6.5 WEIGHT PENALTY

Conversion of the KC-135 brake hydraulic system to the two-fluid configuration will increase the weight of the aircraft between 39 and 64 pounds. The exact increase in weight is dependent upon the diameter of hydraulic lines used in the CTFE portion of the two-fluid system. When the original KC-135 hydraulic line sizes are used (a combination of 1/2 and 3/8 inch diameter lines; see Figure 7) the increase in weight is 39 pounds (see Appendix A for a weight breakdown). However, results of the analytical study of the two-fluid system (Section 2.7) indicate that the hydraulic area of the lines between the deboost valve and brakes should be doubled (increased to 3/4 and 1/2 inch diameter lines; see Figure 7). The larger line sizes would increase the aircraft weight by 64 pounds.

2.7 ANALYSIS OF THE TWO-FLUID BRAKE HYDRAULIC SYSTEM

A dynamic analysis of the KC-135 brake hydraulic system and the two-fluid brake hydraulic system was performed to determine, prior to laboratory testing, (1) the effect which the two-fluid system has on the dynamic response of the brake hydraulic system, and (2) the modifications to the two-fluid system configuration which are required to achieve a dynamic response characteristic comparable to that of the KC-135 brake hydraulic system. The following paragraphs describe the method and results of the analysis.

2.7.1 METHOD OF ANALYSIS

The KC-135 and the two-fluid brake hydraulic systems were analyzed using the government owned Hydraulic System Frequency Response (HSFR) computer program. The brake systems were modelled and HSFR was used to determine the frequency response of each brake system [i.e., the gain and phase angle relationships between the pressure into the antiskid valve (input) and resulting pressure at the brake assembly (output)]. These frequency responses were compared to assess the effects of the two-fluid configuration on the brake system dynamic response.

SYSTEM CONFIGURATION FROM THE ANTISKID VALVE TO THE BRAKE

DESCRIPTION	LINE NUMBER	LINE SIZE (INCHES)	LINE LENGTH (INCHES)
ANTISKID VALVE TO FUSE	1	1/2	11
FUSE TO SHUTTLE	2	1/2	16
DEBOOST TO HOSE	3	1/2	170
HOSE	4	1/2	81
BRAKE LINE	5	3/8	63
BRAKE HOSE	6	3/8	24

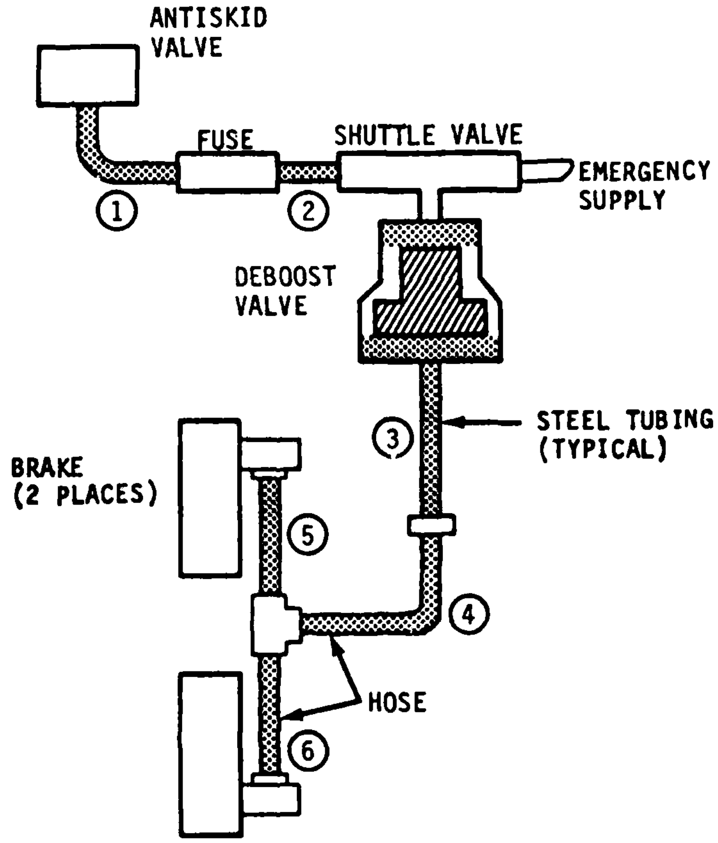


Figure 7 Hydraulic Line Sizes

2.7.2 HSFR COMPUTER MODEL OF THE KC-135 BRAKE HYDRAULIC SYSTEM

The KC-135 brake hydraulic system was modelled from the antiskid valve to the brake. Diagrams of the HSFR KC-135 brake system computer model and the actual KC-135 system are shown in Figure 8. The computer model contains a pump, antiskid valve, hydraulic lines and hoses, deboost valve and brake. The pump models the pilot metering valve and provides a pressure and fluid flow source for the brake system. The effects of the fuse and shuttle valve have been included in the antiskid valve model. A detailed explanation of the computer model, definition of the KC-135 brake hydraulic system configuration and a list of inputs to HSFR model are included in Appendix A.

2.7.3 DYNAMIC RESPONSE OF THE KC-135 BRAKE HYDRAULIC SYSTEM

The computer generated frequency response of the KC-135 brake hydraulic system between 1.0 and 40.0 Hertz is shown in Figure 9. Frequency response is the gain (amplitude) and phase angle (time lag) relationships between an input and output signal as a function of the frequency (sinusoidal oscillation) of the input signal. Gain measured in decibels (db) is a logarithmic ratio of the amplitude of the output signal (i.e., brake pressure) to the amplitude of the input signal (i.e., the pump pressure). Positive gain indicates that the output amplitude is larger than the input amplitude, while negative gain indicates that the output is smaller than the input. Phase angle measured in degrees is an indication of the time delay (or lead) between the peak amplitude of the input and output signals. Negative phase angle indicates that the peak amplitude of the output lags (a time delay) the peak amplitude of the input. Larger negative phase angles (at a particular frequency) indicate more time delay between the input and output.

The performance of modern airplane brake systems is dependent upon the time delay in the brake hydraulic system. For this reason the phase angle relationship is most important. For the purposes of this and following discussions the frequency at -90 degrees phase angle will be called the system breakpoint. A reduction in the breakpoint frequency indicates an increase in the system time delay or lag. The breakpoint frequency of the KC-135 brake hydraulic system as shown in Figure 9 is approximately 8.1 Hertz.

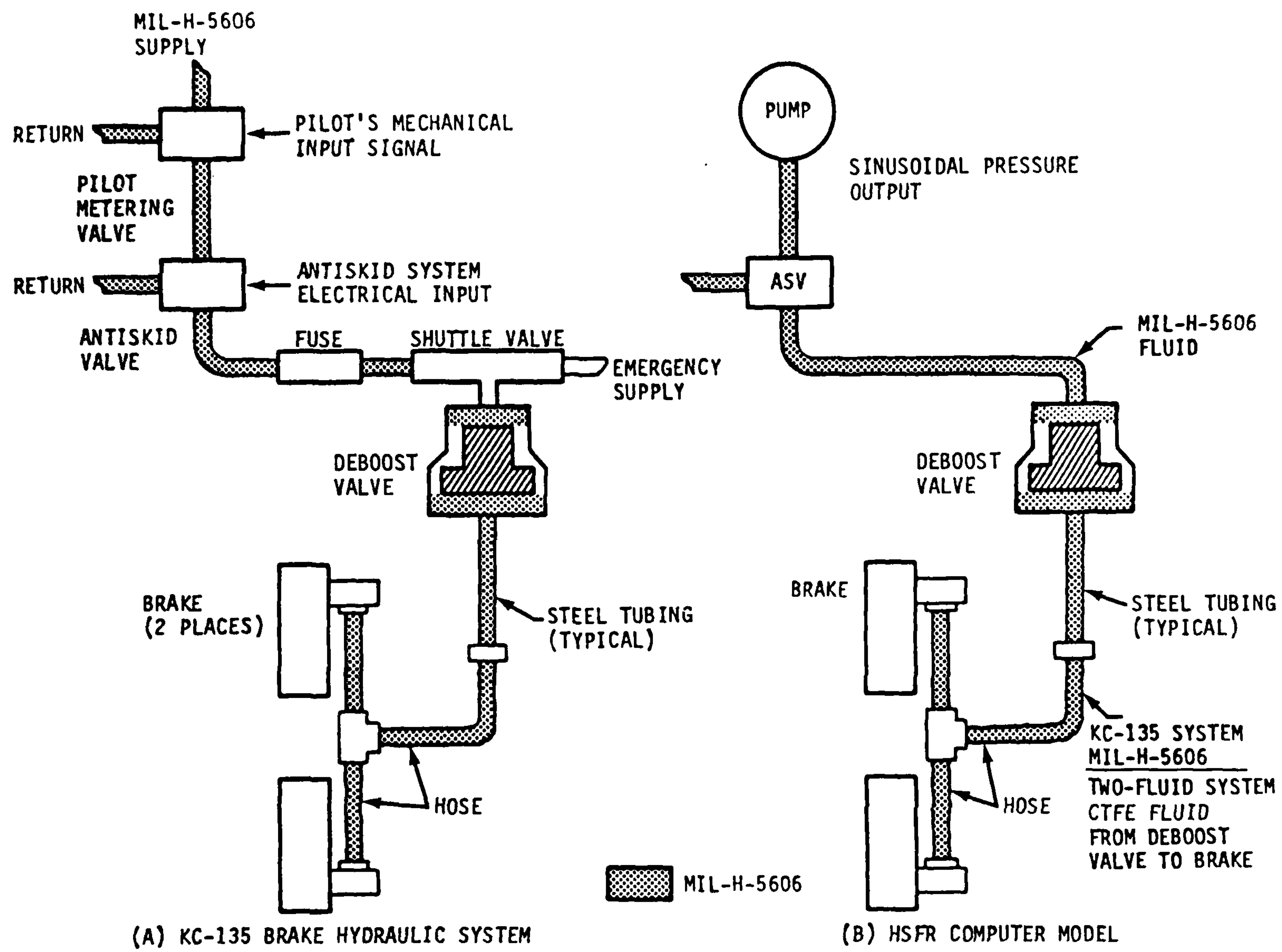


Figure 8 KC-135 Brake Hydraulic System Computer Model

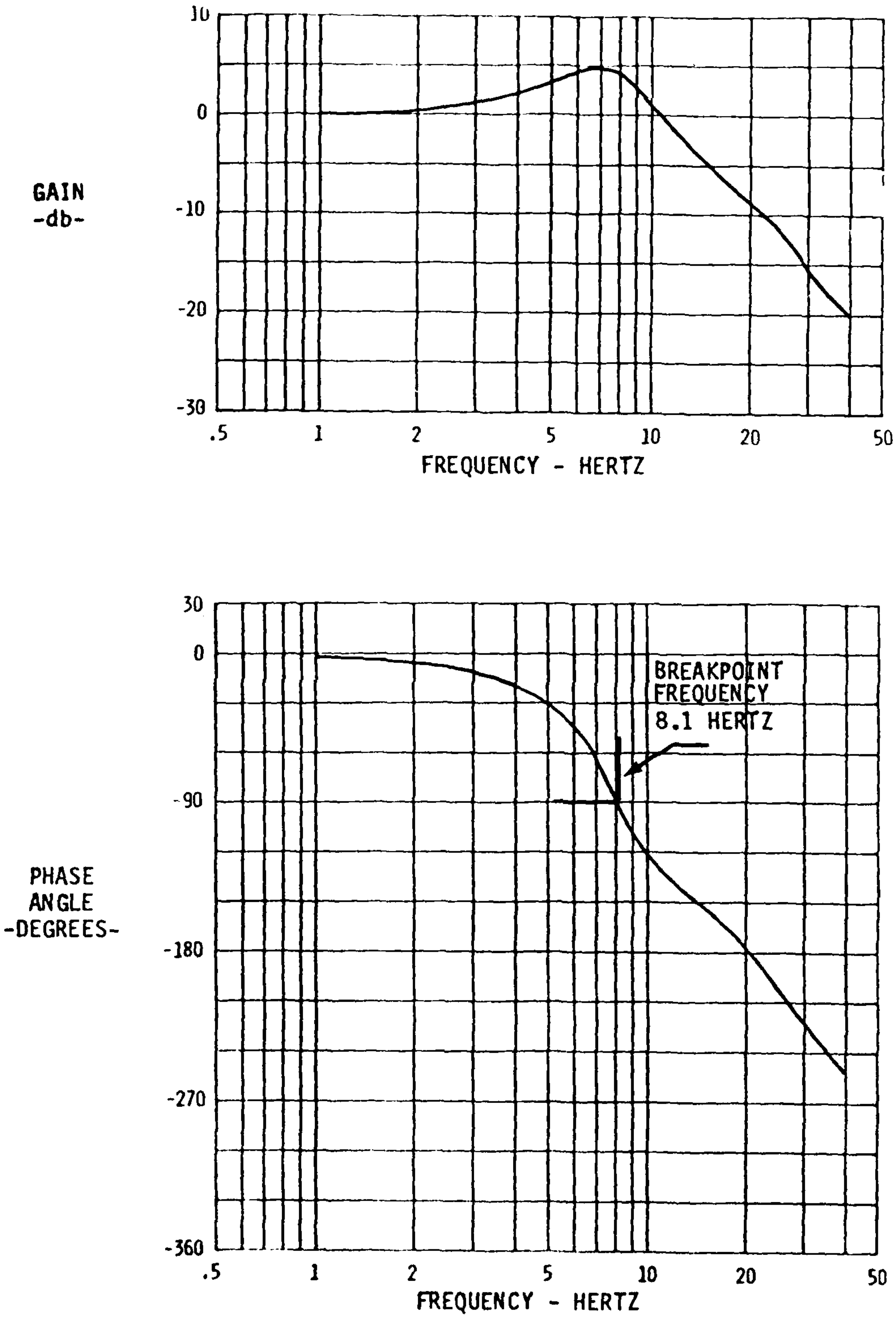


Figure 9 KC-135 Brake Hydraulic System Frequency Response, Computer Model

A series of preliminary computer runs was made to adjust key model parameters and correlate the brake hydraulic system frequency response generated with the computer model with actual laboratory test data (Ref 1). The objective of this correlation effort was to verify the computer model and assure the validity of the predicted results. Excellent correlation between the model and laboratory data was obtained. A complete discussion of the correlation effort and results is given in Appendix A.

2.7.4 TWO-FLUID BRAKE HYDRAULIC SYSTEM DYNAMIC RESPONSE

The KC-135 brake hydraulic system computer model was converted to the two-fluid system configuration and frequency response analyses were performed to determine the effect which the two-fluid system has on the dynamic response of the brake hydraulic system. The two-fluid system configuration is shown in Figure 10. The CTFE replenishment system is functionally inactive during normal operation, and therefore was not modelled for the computer study. The two-fluid system utilizes MIL-H-5606 hydraulic fluid between the pilot's metering valve and the deboost valve and CTFE fluid between the deboost valve and brake, while the KC-135 system uses MIL-H-5606 fluid throughout the entire system. The fluid properties of MIL-H-5606 and CTFE fluids are given in Table 2.

The frequency response of the two-fluid brake system along with the standard KC-135 system response is shown in Figure 11. The computer predicted breakpoint frequency of the two-fluid configuration is 5.6 Hertz. By simply changing from MIL-H-5606 to CTFE fluid between the deboost valve and brake the breakpoint frequency is reduced from 8.1 to 5.6 Hertz. Thus, the CTFE two-fluid brake hydraulic system is slower responding than the standard KC-135 system for frequencies above 2 Hertz.

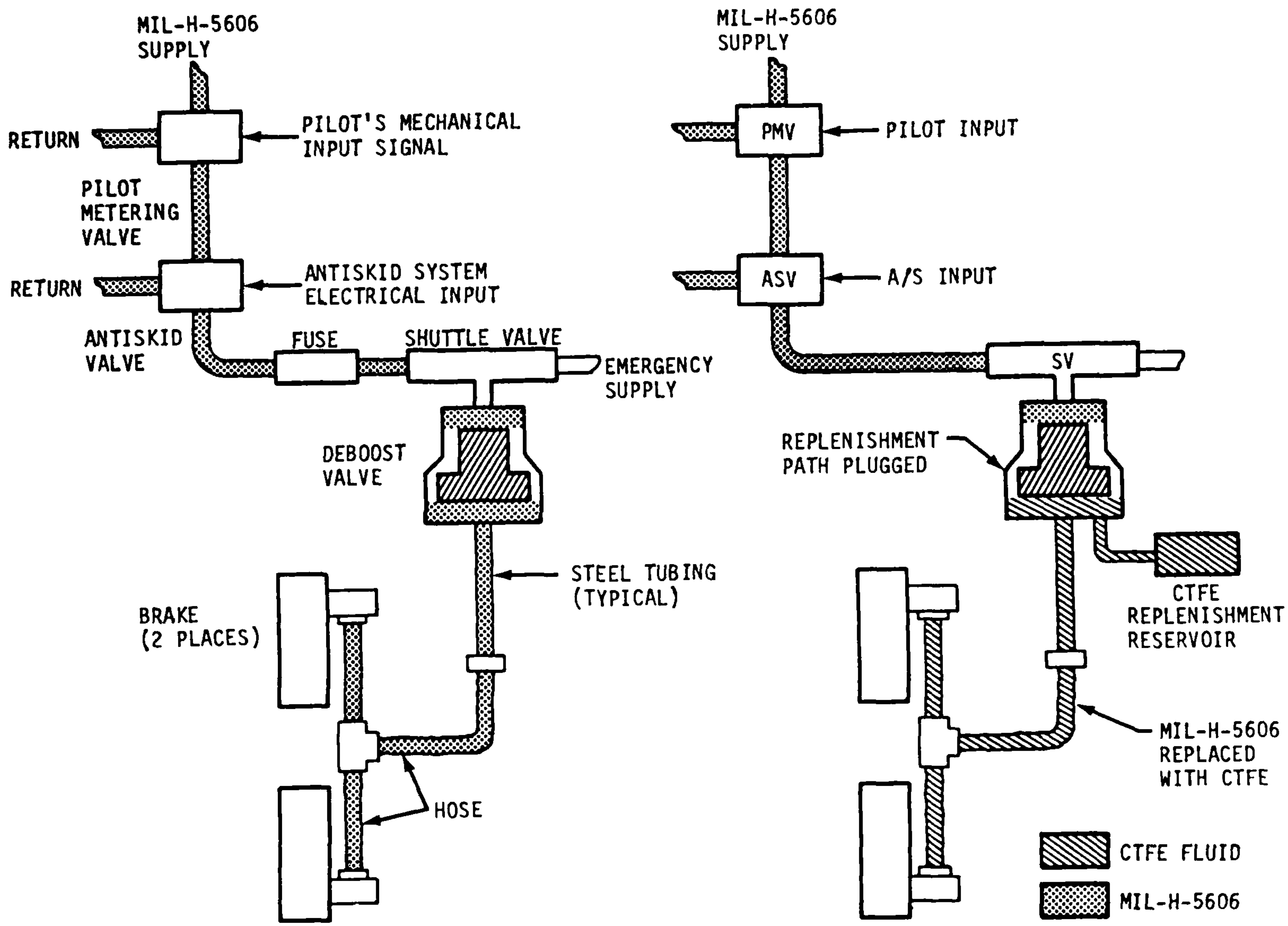
A series of computer runs was made to determine what CTFE fluid property(s) cause the change in system frequency response observed in Figure 11. The fluid properties (bulk modulus, viscosity and density) were varied one at a time to determine the effect each has on the system response. Density was found to be the key parameter responsible for the reduction in the breakpoint frequency. Further discussion and results of this study can be found in Appendix A.

TABLE 2 FLUID PROPERTIES

FLUID PROPERTY	TEMPERATURE (DEGREES F)	MIL-H-5606*	CTFE FLUIDS HALOCARBON AO-8**	CTFE FLUIDS HALOCARBON AO-2**
ADIABATIC	-65	13.47	13.38	13.38
TANGENT BULK	-40	3.25	3.17	3.17
MODULUS X 10^{-5}	0	2.9	2.82	2.82
	50	2.48	2.40	2.40
PSI	100	2.08	2.00	2.00
	150	1.73	1.65	1.65
	200	1.42	1.34	1.34
	250	1.19	1.11	1.11
	300	.98	.90	.90
KINEMATIC	-65	1993.5	2800.0	1100.0
VISCOSITY	-40	482.3	540.0	200.0
	0	134.4	82.0	30.0
	50	34.85	18.7	7.5
CENTISTOKES	100	14.47	7.3	3.1
	150	7.46	3.75	1.72
	200	4.58	2.35	1.08
	250	3.19	1.66	.74
	300	2.39	1.26	.5
VISCOSITY PRESSURE CORRECTION COEFFICIENT	--	.335	.3929	.445
DENSITY X 10^5	-65	8.57	18.70	18.70
LB-SEC^2/IN^4	275	7.63	15.61	15.61

* Data from Air Force HSFR Computer program

** Viscosity data obtained from AFWAL/MLBT. Bulk Modulus and density data obtained from Fire Resistant Aircraft Hydraulic Systems, AFWAL TR-80-2112.



(A) KC-135 BRAKE HYDRAULIC SYSTEM

(B) TWO-FLUID BRAKE HYDRAULIC SYSTEM

Figure 10 Two-Fluid Brake Hydraulic System

2.7.5 SYSTEM CONFIGURATION STUDY

A series of computer runs was performed to determine the two-fluid brake hydraulic system configuration changes which would be required to match the frequency response of the KC-135 system. The only system configuration parameters which can realistically be varied are the antiskid valve gain, flexible hose wall stiffness, and tubing and hose diameters. The analysis showed that replacing the hoses with solid lines (Figure 12) or increasing the line diameters between the deboost valve and brakes (Figure 13) achieves significant improvement in terms of shifting the breakpoint frequency toward the KC-135 response. Changing the antiskid valve gain (fluid flow capacity) does not shift the breakpoint frequency.

While increasing the diameter of the hydraulic lines does shift the system phase angle toward the baseline, it also represents a weight penalty. To match the performance of the baseline system the cross sectional area of the hydraulic lines and hoses must be doubled (i.e., the ratio of fluid density to area must remain constant). Doubling the area along with the 2.11 increase in density (CTFE versus MIL-H-5606) results in over a four fold increase in the weight of the fluid from the deboost valve to the brake. However, the weight penalty can be minimized and still obtain a significant shift of the phase angle toward the baseline by replacing the hoses with solid lines.

2.7.6 STUDY FLUID

Two CTFE fluids, Halocarbon AO-2 and Halocarbon AO-8, were considered for use during the laboratory testing phase of the Fireproof Brake Hydraulic System contract. The physical difference between the two fluids is their kinematic viscosity; their bulk modulus and density are the same (see Table 2). AO-2 is the base stock fluid for AO-8 which is a blend of AO-2 and a viscosity index (VI) improver.

Results of the HSFR computer analysis of the two-fluid system indicated that density is the only fluid property which, when varied significantly affects the dynamic response of the brake system. Changes in kinematic viscosity and bulk modulus have virtually no effect on the system dynamic response.

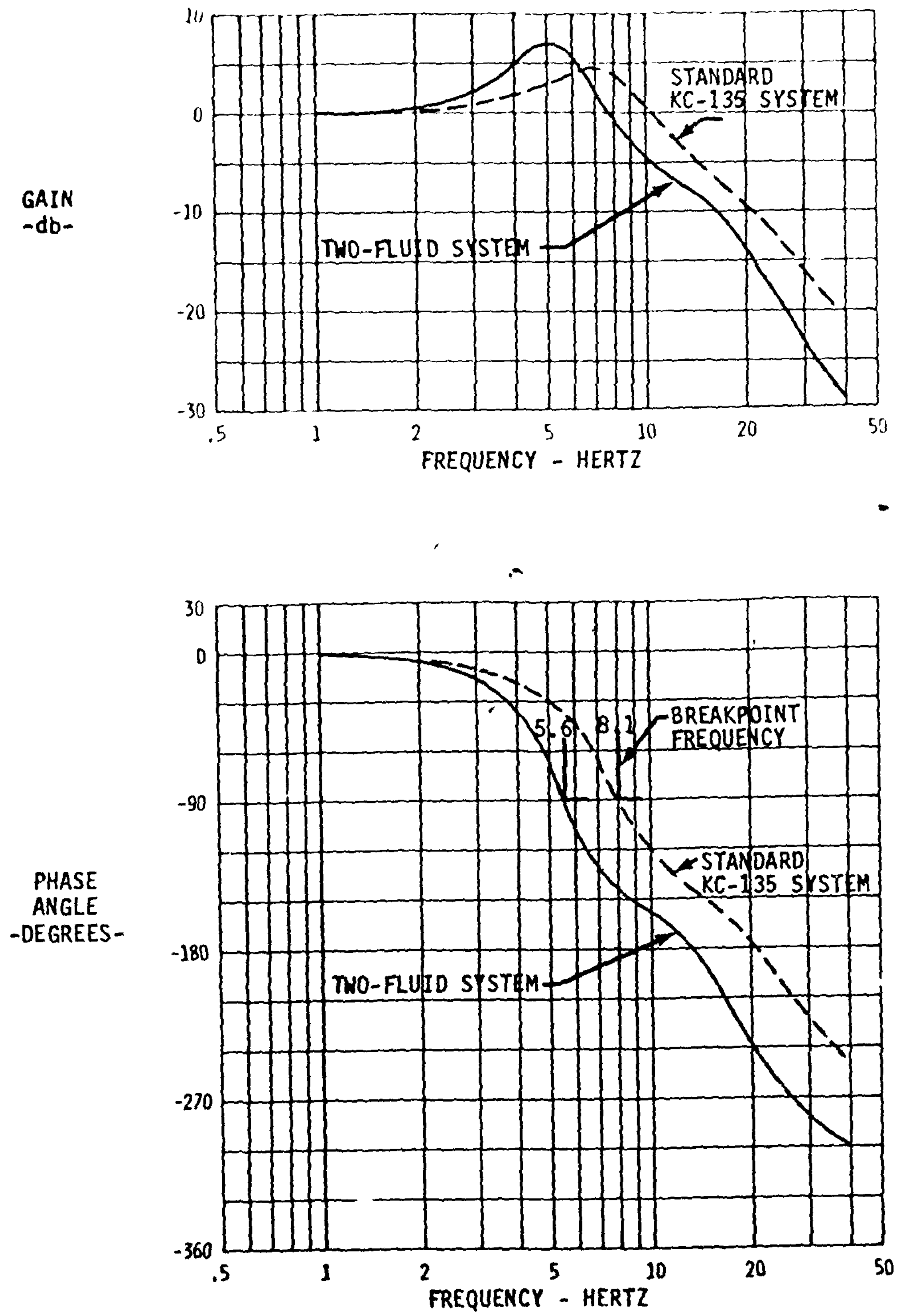


Figure 11 Two-Fluid Brake Hydraulic System Frequency Response

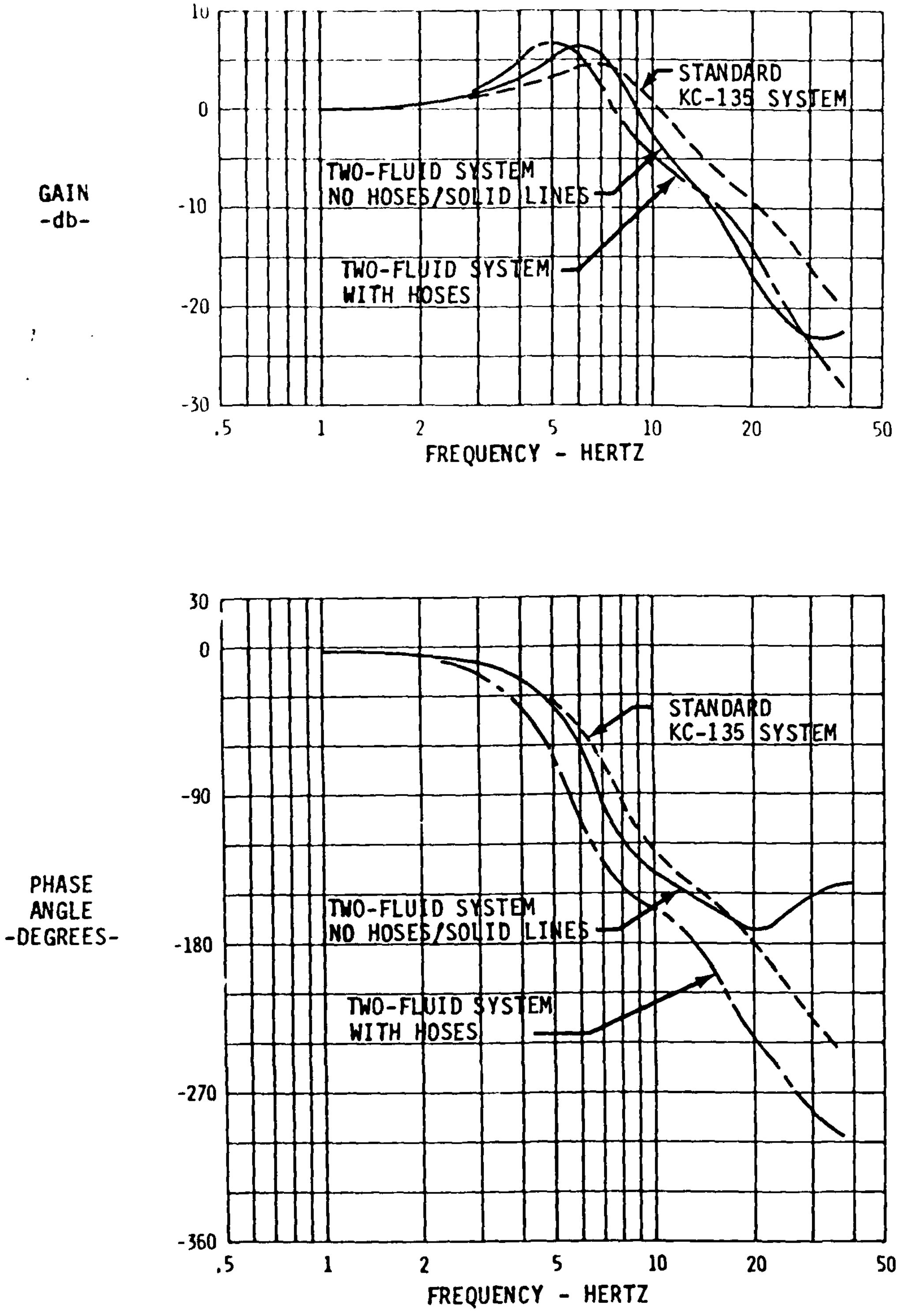
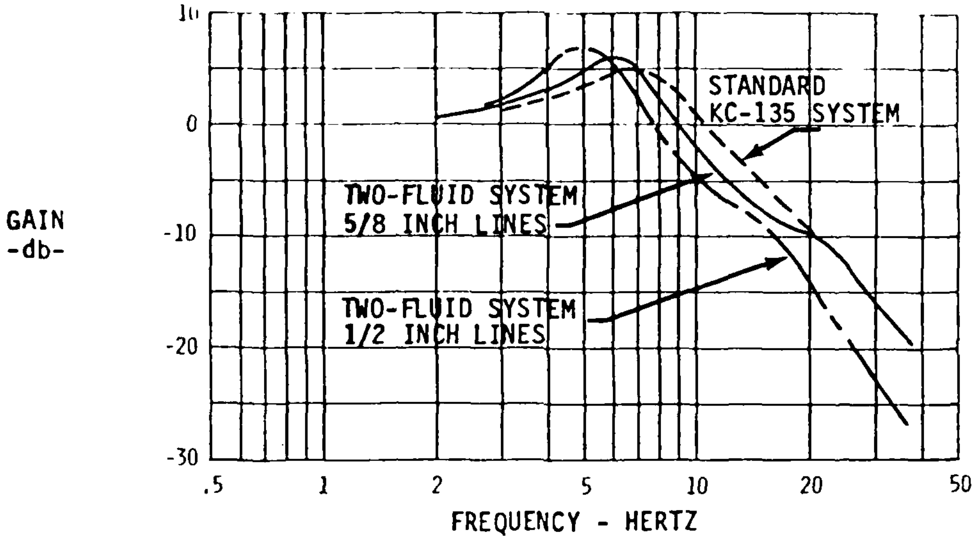


Figure 12 Effect of Hose Elasticity on Frequency Response



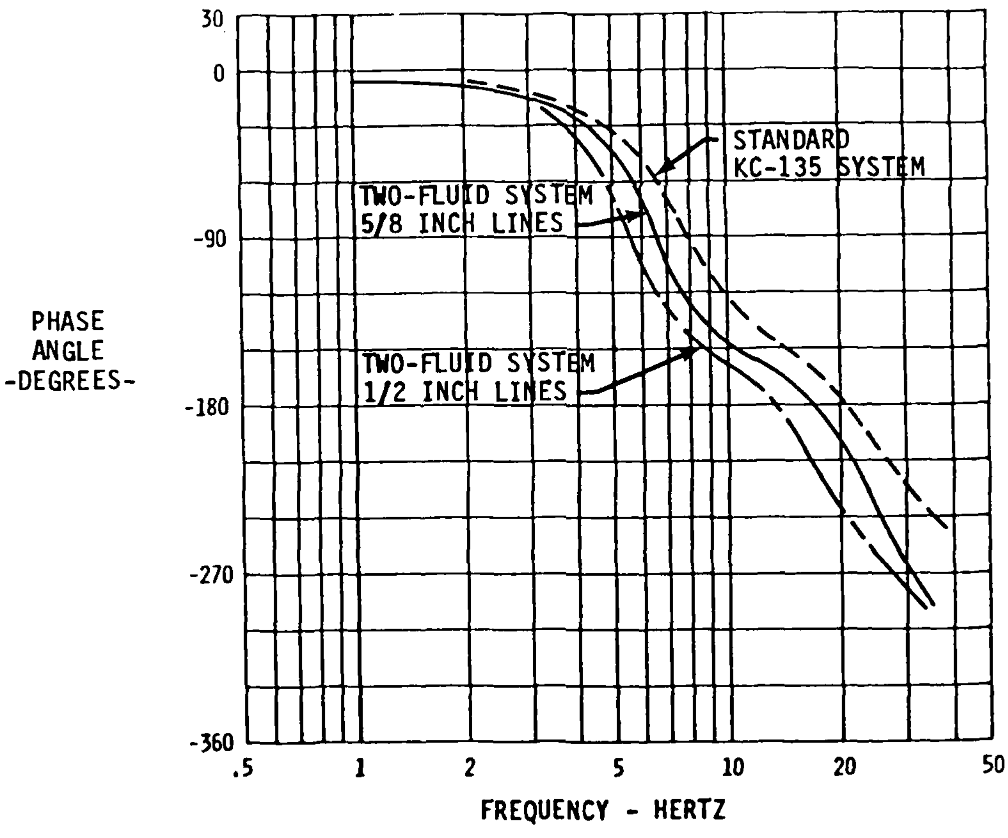


Figure 13 Effect of Line Diameter on Frequency Response

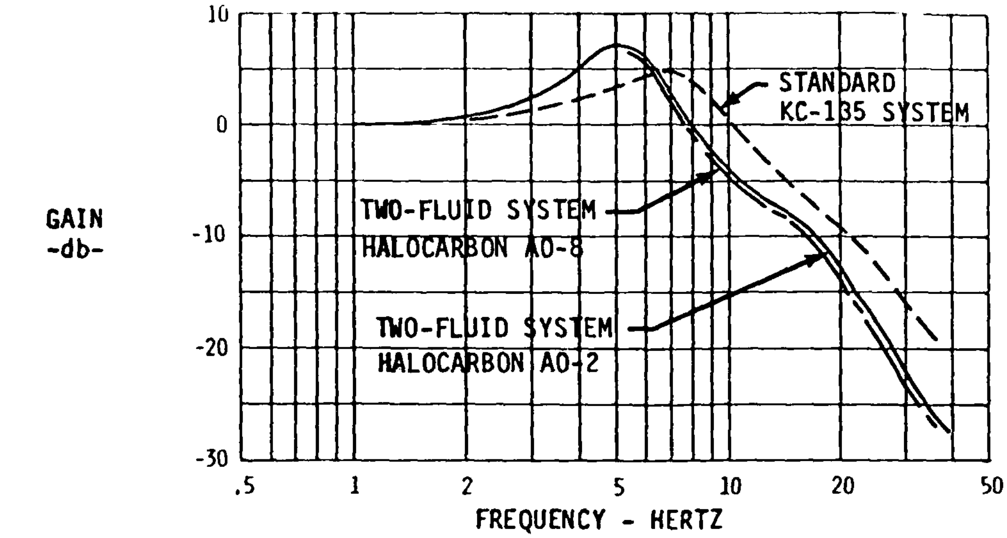
Consequently, the dynamic response of the two-fluid system with AO-8 and with AO-2 are nearly identical, as shown in Figure 14. Since high viscosity is not a requirement for a brake hydraulic fluid, AO-2 was selected for the laboratory test phase. In addition, AO-2 is compatible with MIL-H-5606, whereas AO-8 with the VI improver, when mixed with MIL-H-5606, produces a gel like precipitate (see Section 4.3).

2.7.7 PNF O-RINGS

The PNF seals used during this program were manufactured from PNF-280-001R (80 Durometer) supplied by the Firestone Tire and Rubber Company, Akron, Ohio. This formulation was specified by AFWAL/MLBT for use during the program.

The O-rings were molded by Lord Kinematics, Shelton, Connecticut.

The PNF seals are compatible with both the CTFE and MIL-H-5606 fluids.



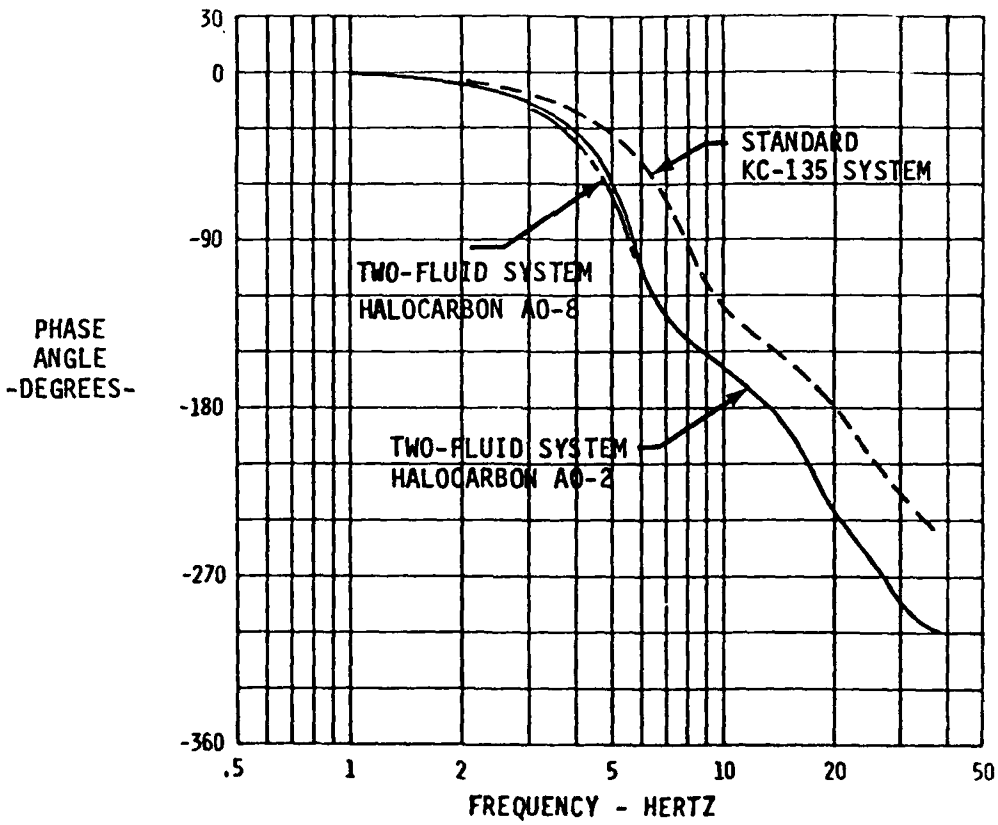


Figure 14 Frequency Response with Halocarbon AO-2 Versus Halocarbon AO-8

SECTION III

LABORATORY TESTING

3.1 TEST OBJECTIVE

A series of laboratory tests were performed to define the performance of components within the two-fluid brake hydraulic system, define the performance of the entire two-fluid brake hydraulic system and determine the affect of the two-fluid configuration on airplane braking performance. The tests were conducted in two parts, component performance tests and system performance tests.

3.2 TWO-FLUID BRAKE HYDRAULIC SYSTEM LABORATORY TEST HARDWARE

The modifications to the KC-135 brake system which are required to form a complete and functional two-fluid brake hydraulic system have been described in Section II. However, to determine the effect of the two-fluid configuration on airplane braking performance only those modifications which are active in the control portion of the brake hydraulic system were made. The CTFE fluid replenishment system does not function during normal braking activity, and therefore was not included in the laboratory test setup.

3.2.1 DEBOOST VALVE MODIFICATIONS

The following modifications were made to the KC-135 deboost valve for the two-fluid brake system laboratory tests.

(1) The original deboost valve replenishment system flow path was plugged.

(2) A new end cap and standpipe assembly was fabricated including the threaded hole for the CTFE replenishment valve.

(3) PNF seals were installed in all areas exposed to CTFE fluid.

The replenishment valve, replenishment reservoir, pilot metered pressure hydraulic line to the reservoir and associated hardware were not fabricated and tested.

A needle valve was installed in the replenishment valve threaded hole so fluid could be pumped through the deboost valve and brake system to accommodate filling and bleeding the CTFE fluid volume. The needle valve was closed during all testing.

The actual hardware modified for testing is shown in Figure 15. A detailed description of the modifications is given in Appendix A. Mechanical drawings of the modifications are included in Appendix B.

3.2.2 BRAKE MODIFICATIONS

The KC-135 five rotor brake assemblies required no design modification for use in the two-fluid brake hydraulic system other than changing to PNF seals. Additional data detailing these modifications and related procedures are given in Appendices A and B.

3.3 COMPONENT PERFORMANCE TESTING

The KC-135 deboost valve and two KC-135 brakes modified for use in the CTFE two-fluid brake hydraulic system mockup were tested to assure that each component met the production part performance requirements prior to its installation and use in the mockup. The component performance tests which were conducted on the deboost valve and brakes are listed in Table 3. The tests were also run on an unmodified KC-135 deboost valve and two standard KC-135 brakes. These tests were performed to provide baseline data which were used to quantitatively determine the effects of the two-fluid modifications on component performance.

Highlights of the component tests performed during the laboratory test phase are given below. A complete description of the test objectives, procedures, hardware, and instrumentation required for each test is given in the Component Performance Test Plan and System Performance Test Plan included in Appendix C. A discussion and compilation of the test results are given in Appendix D.

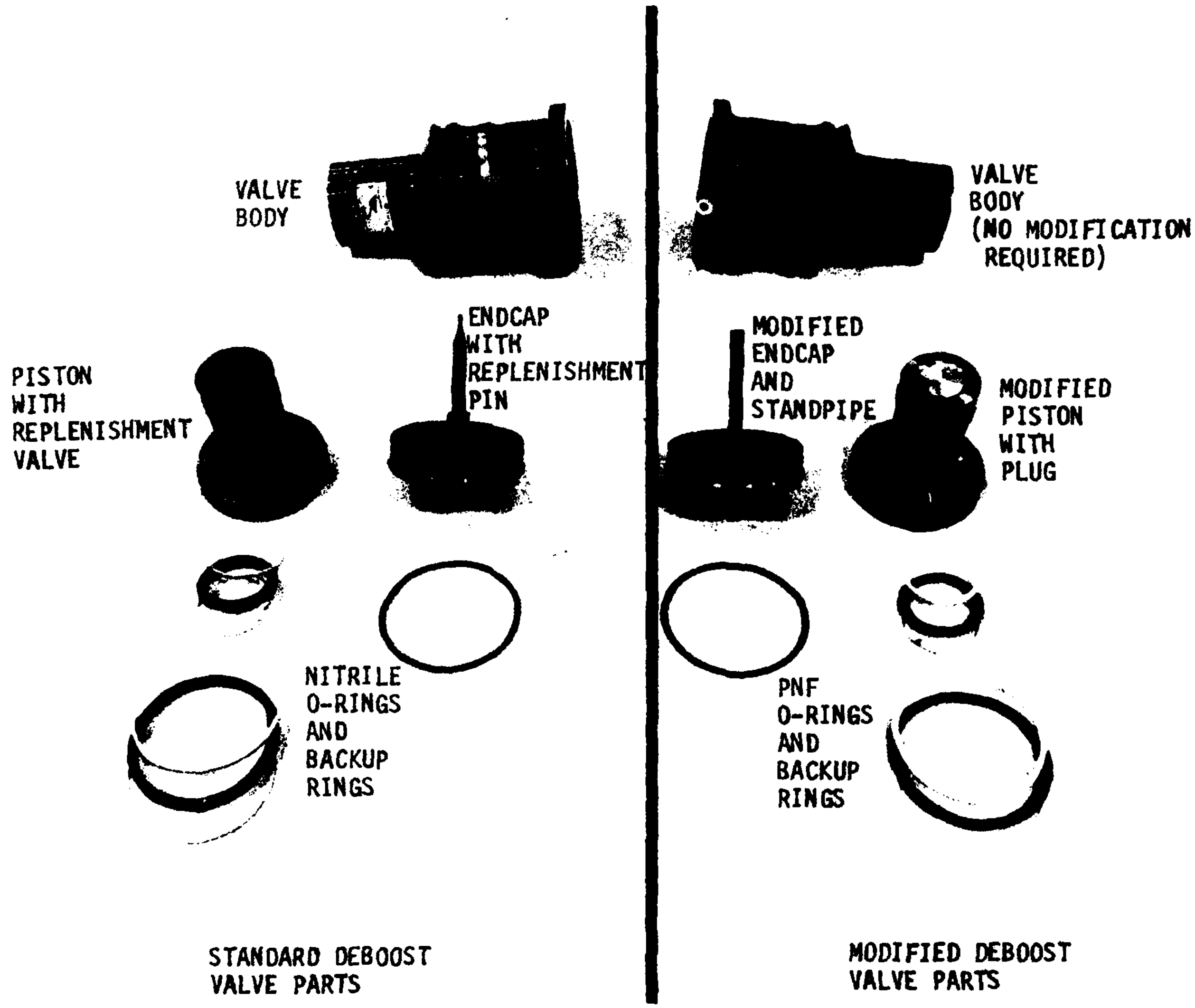


Figure 15 Modified Deboost Valve Test Hardware

TABLE 3 COMPONENT PERFORMANCE TESTS

DEBOOST VALVE COMPONENT PERFORMANCE TESTS

TEST 1	EXAMINATION OF PRODUCT
TEST 2	SEAL BREAK IN
TEST 3	PROOF PRESSURE AND STATIC LEAKAGE
TEST 4	DYNAMIC LEAKAGE
TEST 5	SEAL FRICTION

BRAKE COMPONENT PERFORMANCE TESTS

TEST 1	EXAMINATION OF PRODUCT
TEST 2	SEAL BREAK IN
TEST 3	PROOF PRESSURE AND STATIC LEAKAGE
TEST 4	DYNAMIC LEAKAGE

3.3.1 DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

The modified deboost valve and a standard KC-135 deboost valve were tested to assure that the components functioned properly and met the military and manufacturer's performance requirements for production units.

Before laboratory testing each component was visually inspected (Test 1). The standard KC-135 deboost valve was fully assembled when received from the Air Force. The unit was not disassembled and only a visual inspection of the exterior was performed. No obvious defects were noticed and the valve appeared to be in good operating condition.

The two-fluid deboost valve was assembled using a standard (new) deboost valve body, a modified (new) piston, a custom fabricated end cap, and PNF and nitrile O-rings. All parts were inspected prior to assembly. Each part was in new condition. Assembly of the unit was observed and supervised by the test engineer.

Each deboost valve was subjected to a seal break in period (Test 2) prior to the actual performance tests. During this period the deboost valve piston was cycled up and down to assure proper seating of the new O-ring seals.

Both the modified and standard deboost valve successfully passed all the tests in the component test series. The results of the deboost valve component test are summarized in Table 4. No fluid leakage was observed from either the standard or two-fluid deboost valve during any of the tests. The dynamic leakage and seal friction tests were performed at ambient, +160 degrees Fahrenheit and -65 degrees Fahrenheit which covers the expected temperature range during brake system operation. No leakage or differences in deboost valve operation were observed during the high and low temperature tests.

The frictional force of deboost valve piston dynamic seals was measured during the seal friction test (Test 6). The results are given in Table 4. The magnitudes of the frictional force in both the modified and standard deboost valves at ambient and high temperatures are significantly larger than the frictional force at low temperatures. The friction in both units is reduced

TABLE 4. DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEST CONDITIONS TEMP.	RESULTS AND COMMENTS STANDARD	RESULTS AND COMMENTS TWO-FLUID
1. Examination of Product	—	—	Assembly Inspected Visually, Not Disassembled To Inspect Internal Parts	Manufactured and Modified Parts Visually Inspected, Assembly Observed
2. Seal Break In	50 Cycles, 0-300 psi 50 Cycles, 0-600 psi 100 Cycles, 0-965 psi	Ambient	No Leakage	No Leakage
3. Proof Pressure and High Pressure Seal Static Leakage	4500 psi at Port A for 2 Minutes	Ambient	No Leakage	No Leakage
4. Proof Pressure and High Pressure Seal Static Leakage	1445 psi at Port B for 2 Minutes	Ambient	No Leakage	No Leakage
5. Dynamic Leakage				
5a Ambient	25 Cycles, 0-965 psi	Ambient	No Leakage	No Leakage
5b -65°F		-65°F	No Leakage	No Leakage
5c 160°F		160°F	No Leakage	No Leakage
6. Seal Friction				
6a Ambient	1500 psi at Port A	Ambient	No Leakage Seal Friction -102 lb	No Leakage Seal Friction -90 lb
6b -65°F		-65°F	No Leakage Seal Friction 4 lb	No Leakage Seal Friction -2 lb
6c 160°F		160°F	No Leakage Seal Friction -27 lb	No Leakage Seal Friction -69 lb

to nearly zero at low temperature. The loss of friction at low temperature is most likely due to the difference in the thermal expansion (contraction) coefficients of the aluminium deboost valve piston and body and the PNF (or nitrile) O-rings. At low temperature this difference results in less seal compression and hense less friction.

A CTFE fluid sample taken at the conclusion of the modified deboost valve testing showed no obvious discoloration (e.g., reddishness due to mixing of CTFE and MIL-H-5606) indicating that the modified deboost valve successfully isolated the CTFE and MIL-H-5606 fluids.

A complete discussion of the deboost valve component tests and detailed results are given in Appendix D.

3.3.2 BRAKE COMPONENT PERFORMANCE TEST RESULTS

Two standard KC-135 brakes and two modified brakes were tested to assure that each component functioned properly and met the military and manufacturer's performance requirements for production units. Each brake unit successfully passed all the brake component performance tests. The results of these tests are summarized in Table 5.

Both the standard and CTFE fluid brakes were modified prior to testing. The CTFE fluid brakes were rebuilt using new brake pistons, new piston bushings and PNF O-rings. Each brake part was cleaned and visually inspected (Test 1) prior to reassembly. All the replacement and cleaned parts were in new condition. Reassembly of the brake was observed and supervised by the test engineer. The KC-135 brakes obtained from the Air Force utilized a T-seal for the dynamic piston seal. These T-seals were replaced with a standard nitrile O-ring. During this replacement the standard brakes were inspected (Test 1). The brake pistons and piston bushings were in excellent condition and showed no signs of wear.

Prior to the performance tests each brake was subjected to a seal break in period (Test 2). This test was designed to assure proper seating of dynamic

TABLE 5. BRAKE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEST CONDITIONS TEMP.	RESULTS AND COMMENTS STANDARD BRAKE	RESULTS AND COMMENTS CTFE FLUID BRAKES
1. Examination Of Product	—	—	Replaced T-Seal With O-Ring No Wear on Piston or Bushings	Cleaned and Inspected all Parts, Replaced T-Seal With PNF O-Rings
2. Seal Break In	50 Cycles, 0-300 psi 50 Cycles, 0-600 psi 100 Cycles, 0-900 psi	Ambient	No Leakage	No Leakage
3. Proof Pressure Static Leakage	1800 psi For 5 Minutes	Ambient	No Leakage	No Leakage
4. Dynamic Leakage				
4a Ambient	25 Cycles, 0-1200 psi	Ambient	No Leakage	No Leakage
4b -65°F		-65°F	No Leakage Slow Piston Release	No Leakage Slow Piston Release
4c 160°F		160°F	No Leakage	No Leakage

seals prior to performance testing (Tests 3 and 4). The brakes operated normally during all tests. No fluid leakage was observed.

The brake pressure and brake piston displacement were monitored during the dynamic leakage test (Test 4). The brakes were pressurized to the maximum operating pressure (1200 psi) then a step decrease in brake pressure was commanded. Typical pressure decay and piston retraction time history plots for a standard brake and a CTFE fluid brake are shown in Figure 16. The pressure in each brake drops rapidly while the brake piston takes a somewhat longer time to retract. The piston displacement retraction time of the two-fluid brake at -65 degrees Fahrenheit is approximately 0.2 second faster than the retraction time of the standard brake although the pressure decay time histories are nearly the same.

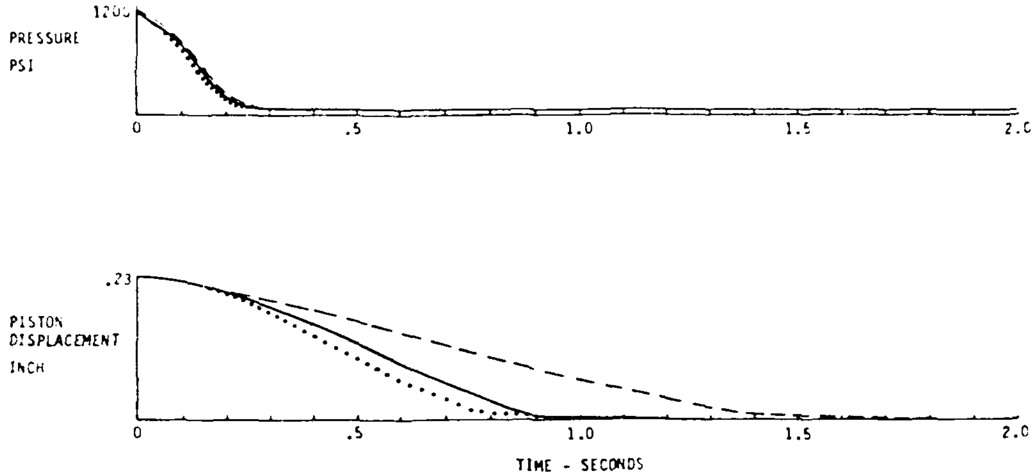
3.4 SYSTEM PERFORMANCE TESTING

The system performance tests listed in Table 6 were conducted to determine: (1) the operational characteristics and dynamic response of the brake hydraulic system and specific components within the system, and (2) the stopping performance of the brake system. The tests were performed using a mockup of the two-fluid brake hydraulic system (Figure 17) and a mockup of the KC-135 brake hydraulic system (Figure 18). The results of the tests were compared to determine the effect of the CTFE two-fluid brake system configuration on brake system dynamic response and airplane stopping performance. A complete description of each test, the test procedures and the test conditions is given in Appendix C.

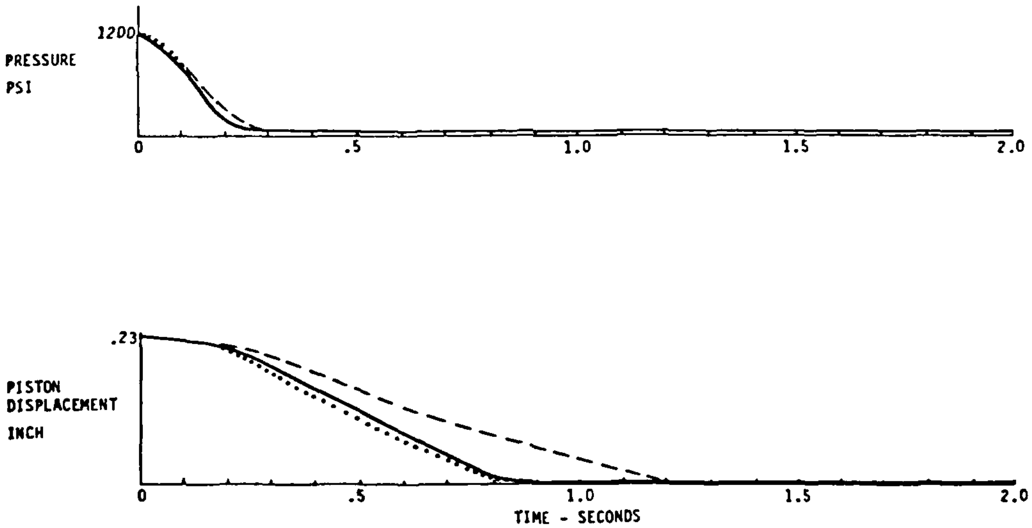
A complete description and compilation of the test results are given in Appendix E. A summary of these results are presented in the following paragraphs in two sections.

3.4.1 EFFECT OF THE TWO-FLUID CONFIGURATION ON DYNAMIC RESPONSE

Frequency response (Test 1) and step response (Test 2) tests were performed on both the two-fluid brake hydraulic system mockup and the standard KC-135 brake hydraulic system mockup. The results of these tests were compared to



(a) STANDARD KC-135 BRAKE, NITRILE SEALS-MIL-H-5606 FLUID



(b) CTFE FLUID BRAKE, PNF SEALS-CTFE FLUID

Figure 16 Brake Piston Displacement

TABLE 6 SYSTEM PERFORMANCE TESTS

OPERATIONAL CHARACTERISTICS AND DYNAMIC RESPONSE TESTS

TEST 1. FREQUENCY RESPONSE

TEST 2. STEP RESPONSE

TEST 3. STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE

TEST 4. STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME

STOPPING PERFORMANCE TESTS

TEST 5. CONSTANT FRICTION RUNWAY

TEST 6. WET RUNWAY

TEST 7. STEP FRICTION

TEST 8. LANDING GEAR SYSTEM STABILITY

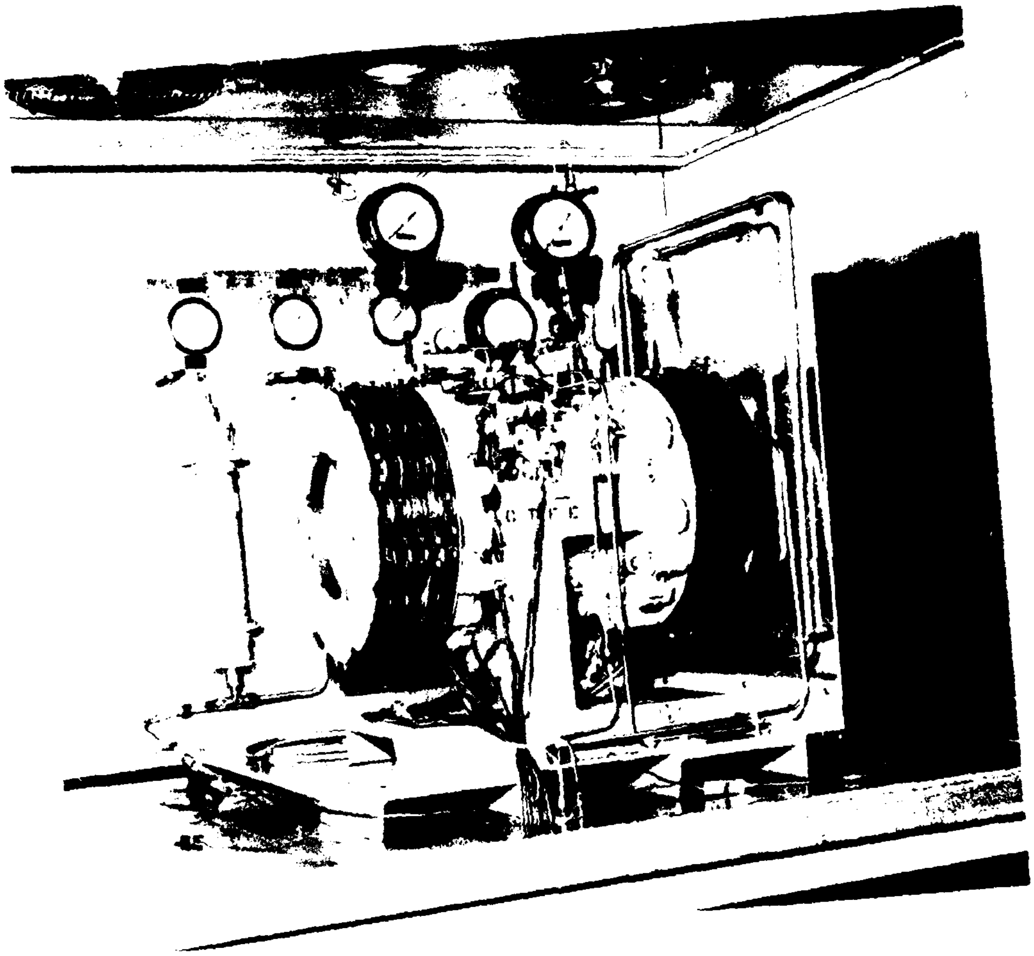


Figure 17 Two-Fluid Brake Hydraulic System Mockup

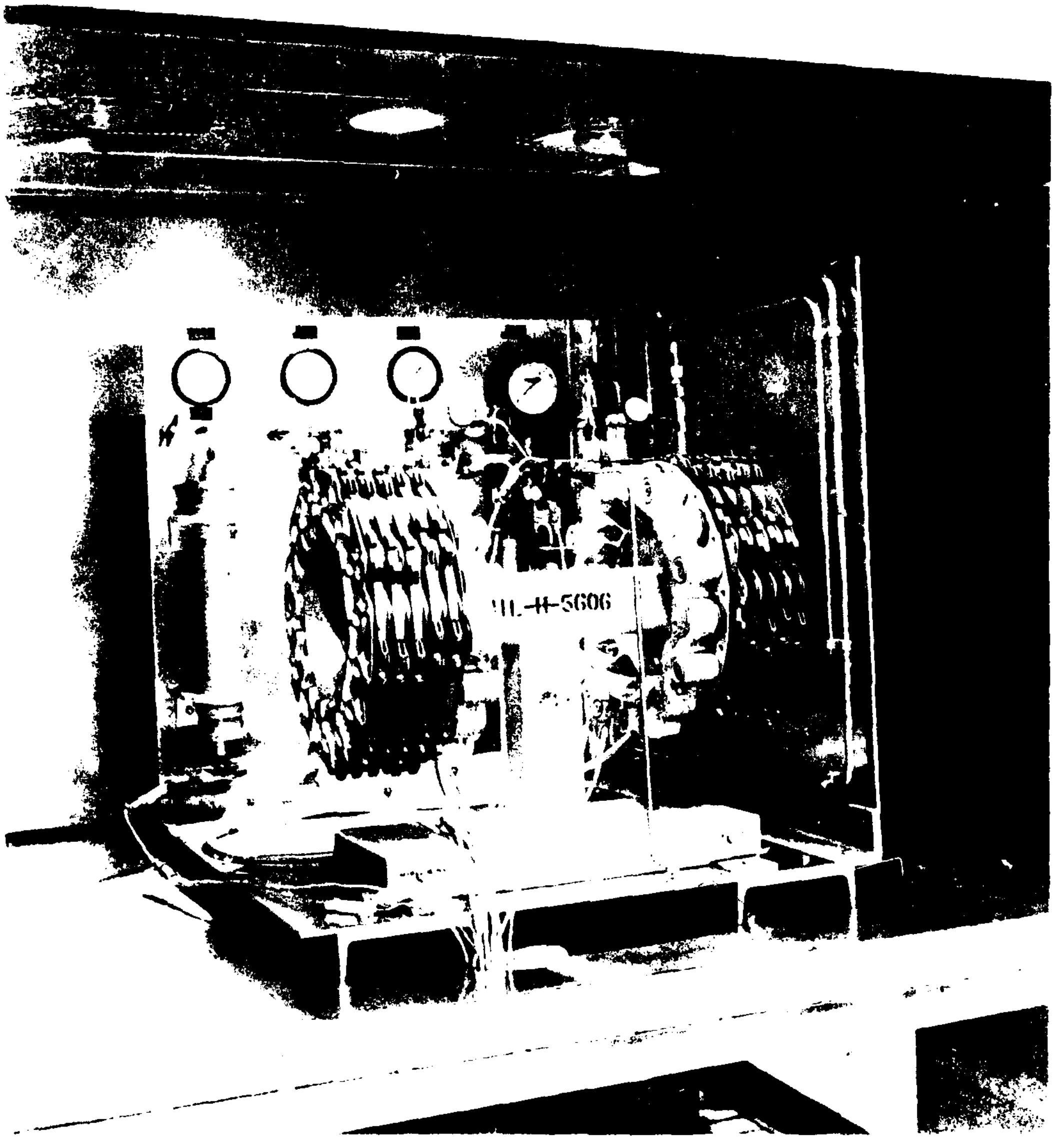


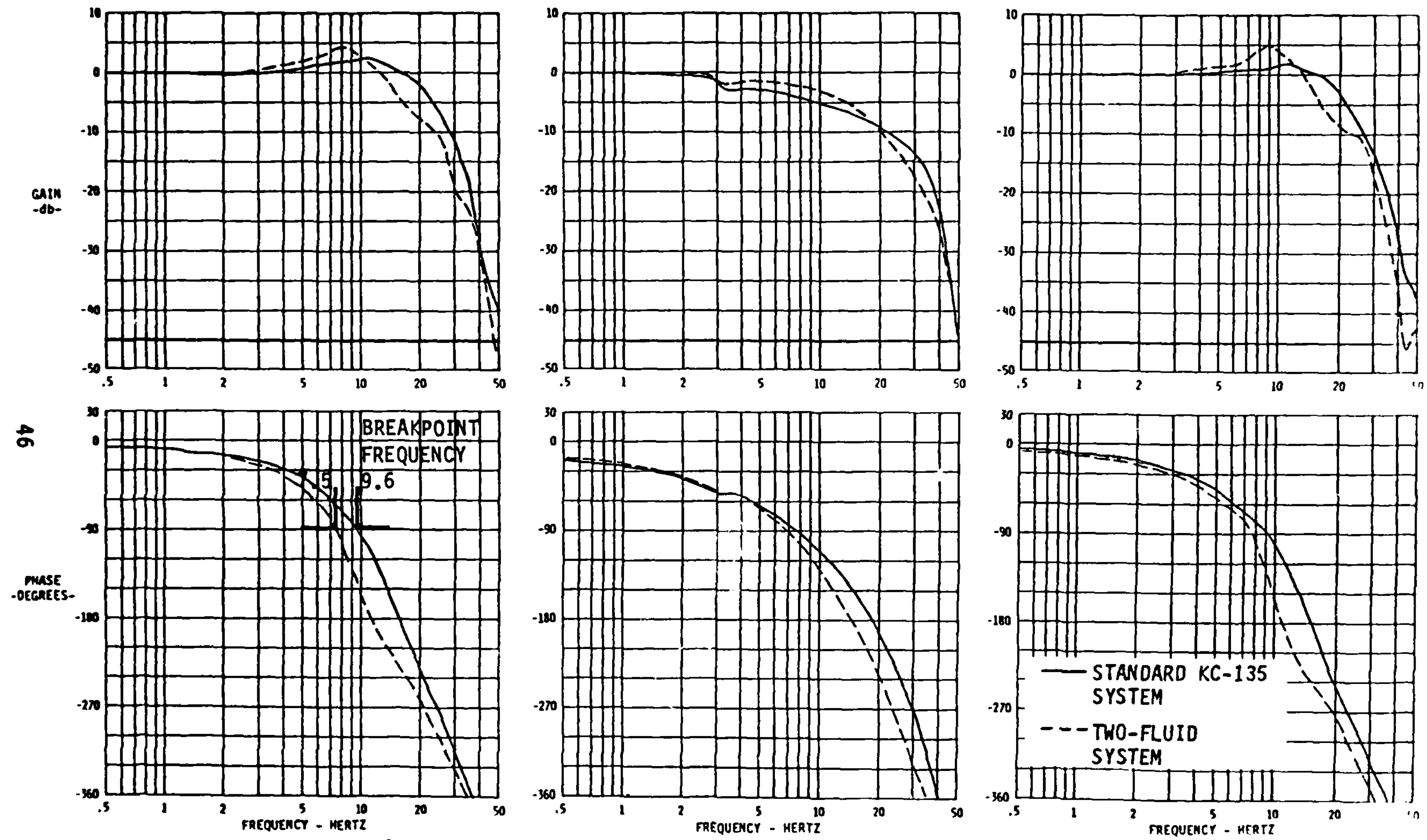
Figure 18 KC-135 Brake Hydraulic System Mockup

determine the effect of the two-fluid configuration on the dynamic response of the deboost valve, the antiskid valve and the entire brake hydraulic system. Frequency response curves generated at three temperatures (ambient, -40°F and 160°F) are shown in Figures 19, 20 and 21. The effects on the entire brake hydraulic system from the antiskid valve to the brake (Figure 2) are shown in Figure 19. The effects on the antiskid valve alone are shown in Figure 20 and the effects on the deboost valve alone are shown in Figure 21.

The breakpoint frequency of the two-fluid brake system is approximately 7.5 Hertz while the breakpoint frequency of the standard KC-135 brake system is 9.6 Hertz (Figure 19). The reduction of the breakpoint frequency indicates that the response of the brake system has decreased. Consequently, the two-fluid brake hydraulic system is slower than the standard system for dynamic system responses above 2 Hertz.

Although the phase angle characteristics of the standard versus two-fluid antiskid valve and the standard versus two-fluid deboost valve are different, the breakpoint frequency of each component is not significantly affected. Since the breakpoint frequency is not shifted the primary dynamic response characteristics of the deboost valve and antiskid valve components are not changed by conversion to the two-fluid configuration. The shift in the phase angle characteristics of the antiskid valve and deboost valve are second order effects caused by the increased density of the CTFE fluid.

The reduction in brake hydraulic system breakpoint frequency correlates well with the predicted results of the HSFR computer analysis (Section 2.7.4 and Figure 11). In addition, the increased gain of the two-fluid configuration was accurately predicted. The correlation between the values of the predicted and measured breakpoint frequencies are not exact. The discrepency is due to a difference in the stiffness of the hoses used during these tests and those used in the tests (Reference 1) with which the computer model was correlated. The earlier tests used reinforced rubber hose while a steel jacket reinforced rubber hose was used during the current tests. The increased stiffness of the steel jacket hose has caused an increase in the breakpoint frequency (i.e., the breakpoint frequency is proportional to the square root of stiffness).

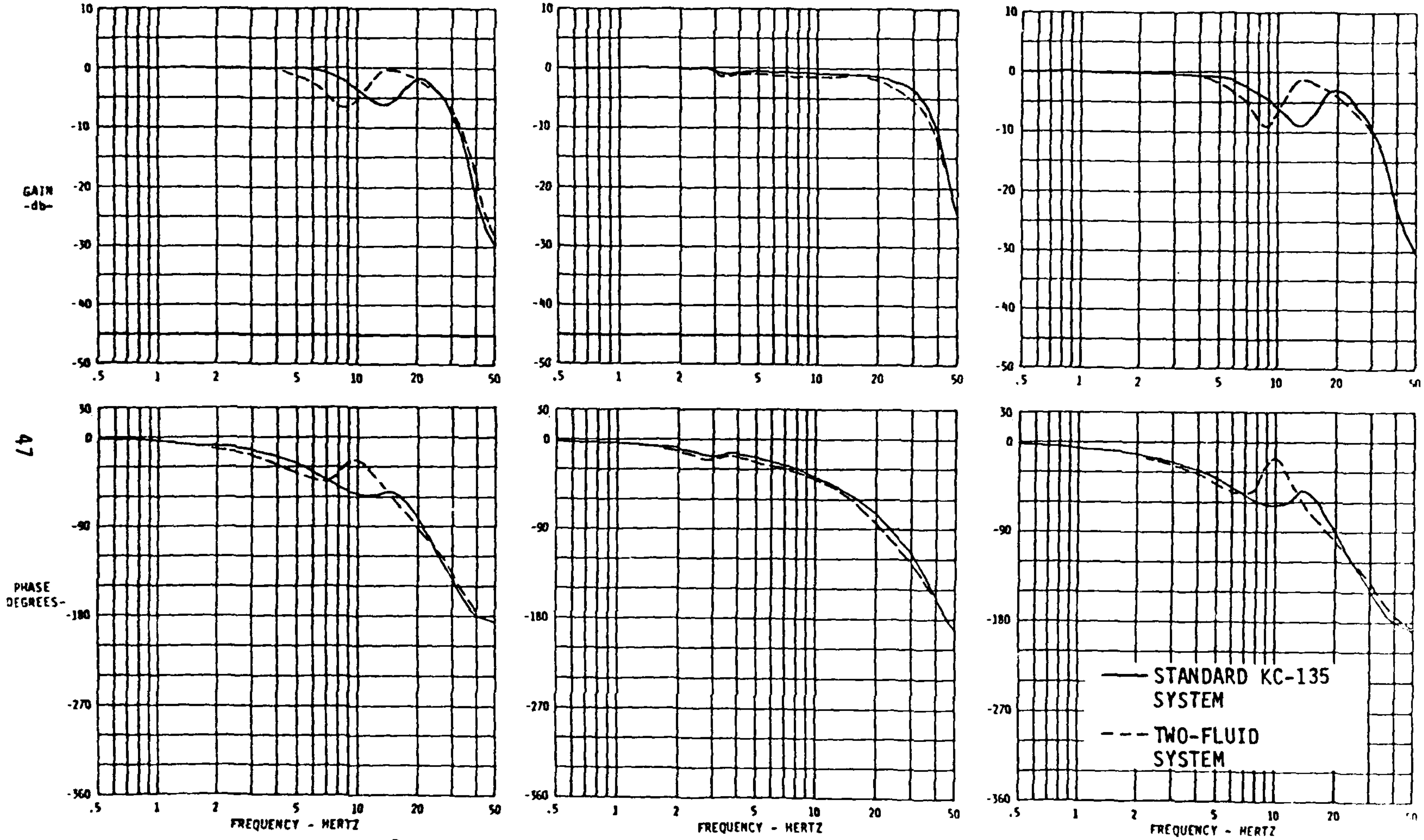


(a) AMBIENT (APPROX. 70°F)

(b) -40° FAHRENHEIT

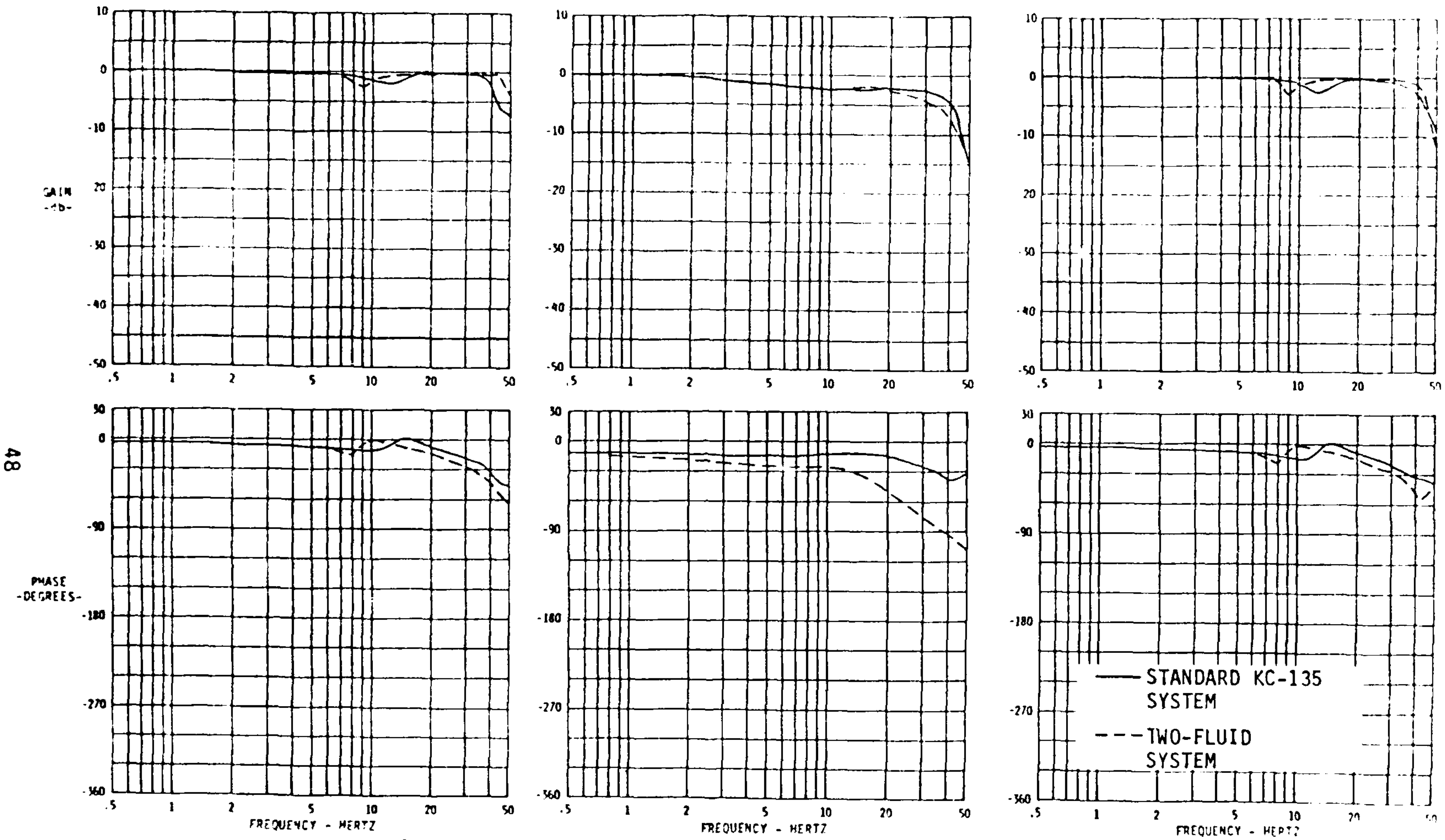
(c) 160° FAHRENHEIT

Figure 19 Brake System Frequency Response



(a) AMBIENT (APPROX. 70°F) (b) -40° FAHRENHEIT (c) 160° FAHRENHEIT

Figure 20 Antiskid Valve Frequency Response



(a) AMBIENT (APPROX. 70°F)

(b) -40° FAHRENHEIT

(c) 160° FAHRENHEIT

Figure 21 Deboost Valve Frequency Response

Results of the step response test (Test 2) also indicate that the dynamic response of the two-fluid brake system is slower than the standard KC-135 brake system. The step response of the two-fluid brake hydraulic system and the standard KC-135 brake system are compared at three temperatures (ambient, -40°F and 160°F) in Figures 22 and 23. The time history data shown in Figure 22 are plots of hydraulic pressure measured at the brake in response to a commanded step increase in pressure occurring at reference time zero. The plots show that the two-fluid system requires more time to pressurize the brake. For example, at ambient temperature the two-fluid system requires 0.03 second longer to reach 500 psi than the standard hydraulic system. The decreasing pressure step response measured at the brake is shown in Figure 23. The time history plots show that the two-fluid brake system requires nearly the same time to dump brake pressure from 500 to 0 psi as does the standard system. The 0 to 500 psi pressure step response was selected for display here because the primary operating mode of the KC-135 antiskid brake system is to step pressure on and off in this pressure range. Consequently, the two-fluid system exhibits slower response in its primary mode of operation.

3.4.2 EFFECT OF THE TWO-FLUID CONFIGURATION ON STOPPING PERFORMANCE

The impact of the two-fluid brake hydraulic system configuration on airplane stopping performance was assessed by comparing the stopping distance of the two-fluid system with the stopping distance of the standard KC-135 brake system. The Boeing Hybrid Brake Control Laboratory (described in Appendix F), the KC-135 airplane computer model, the KC-135 Mark II antiskid control box and the brake hydraulic system mockups (Figures 17 and 18) were combined to form a KC-135 airplane braking computer simulation (discribed in Appendix G). The simulation was used to determine aircraft stopping distance from brake application at approximately 200 feet per second forward airplane velocity to 24 feet per second, subject to a variety of runway conditions.

The stopping distance of the two-fluid and standard KC-135 brake system as a function of the runway friction coefficient at three temperatures (ambient, -40 degrees Fahrenheit and 160 degrees Fahrenheit) is shown in Figure 24. These data are also presented in tabular form in Table 7.

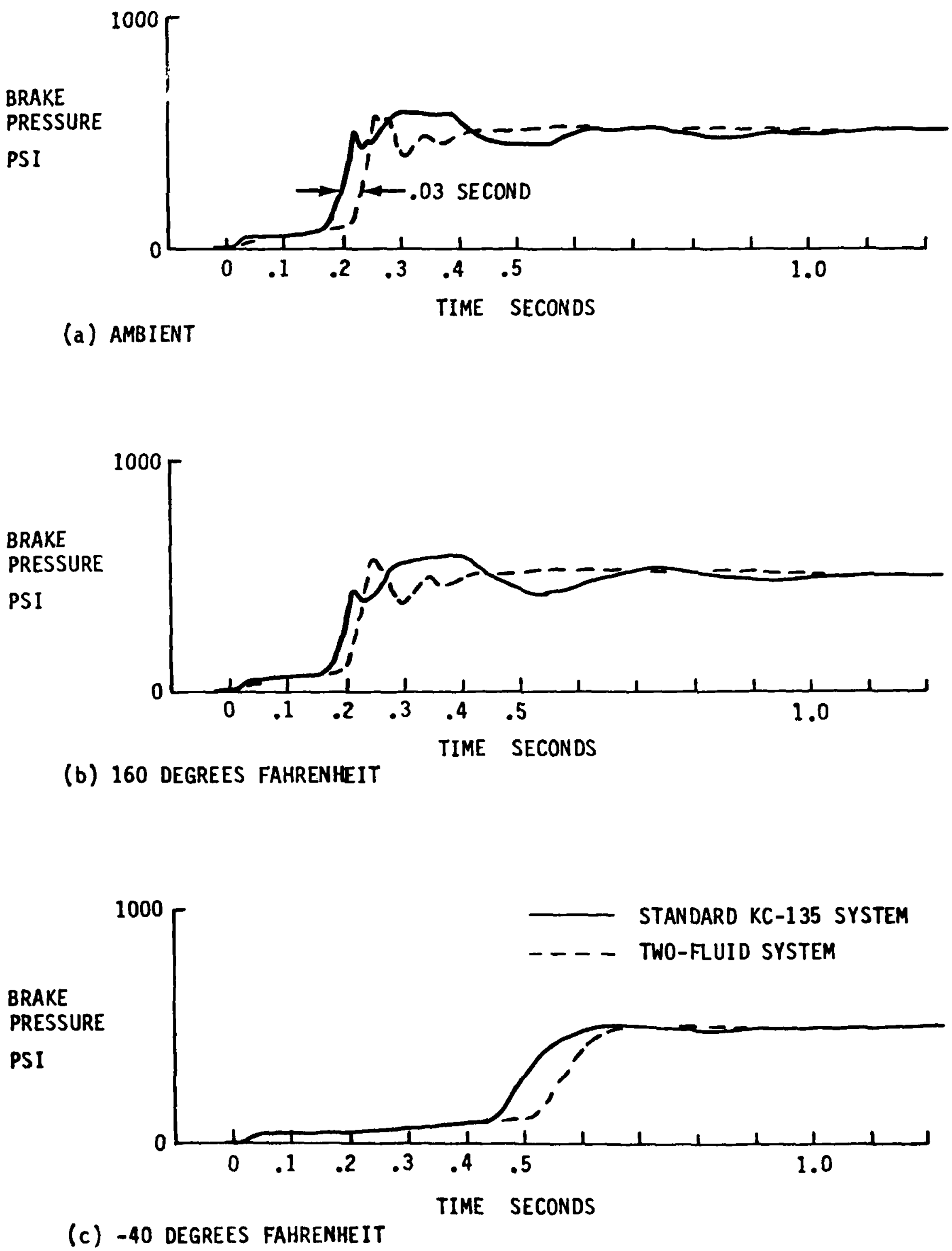


Figure 22 Brake System Step Response, Increasing Pressure Step

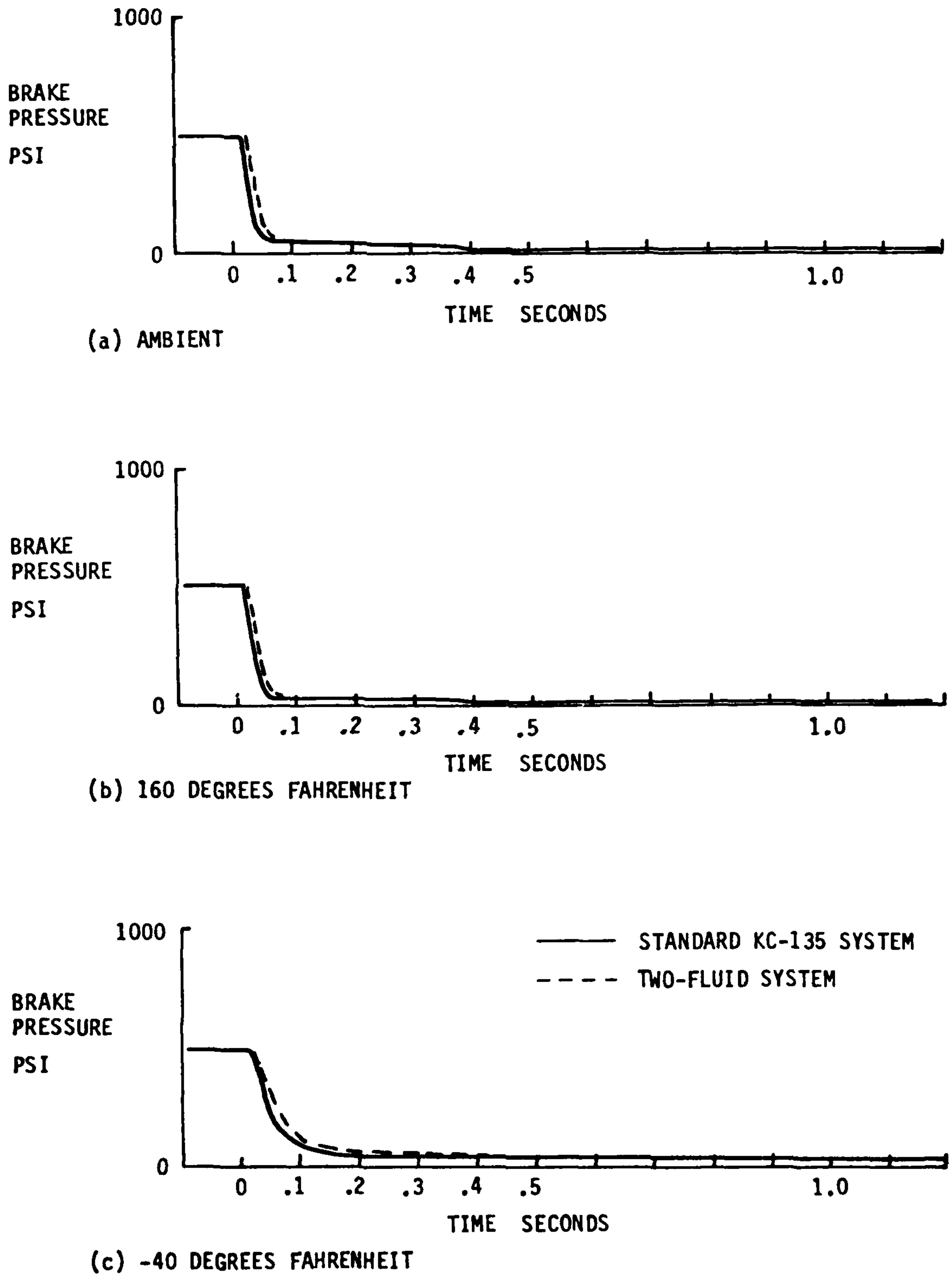


Figure 23 Brake System Step Response, Decreasing Pressure Step

The two-fluid system generally requires more distance than the standard brake system to stop the airplane. This increase in stopping distance is directly attributed to the slower dynamic response of the two-fluid brake hydraulic system (Section 3.4.1). The following description of antiskid brake control system operation is included here to help explain the increase in stopping distance associated with the two-fluid brake hydaulic system.

The primary purpose of the antiskid brake control system is to prevent wheel skidding while maximizing braking performance. The basic control characteristics of the standard KC-135 brake system can be seen in the wheel speed, brake pressure and antiskid valve current signal time history plots shown in Figure 25. A block diagram of the brake control system including the basic features of the antiskid system is given in Figure 26 to aid in the discussion. When a wheel skid occurs (i.e., a rapid decrease in wheel speed caused by excessive brake torque) it is sensed in the antiskid control box via the wheel speed transducer signal sent to the velocity amplifier. A velocity signal generated in the velocity amplifier is sent to the deceleration amplifier where it is differentiated to produce a wheel deceleration signal. This deceleration signal is then compared to a deceleration threshold value. When this deceleration threshold is exceeded the control system immediately requests a full brake pressure dump (see Figure 25). During the pressure dump brake torque is relieved, the wheel spins up and the antiskid system reapplies brake pressure. The pressure is quickly reapplied to a level slightly lower than the pressure at which the skid occurred. The pressure is then ramped on slowly by the pressure bias modulation unit (PBM) until another skid occurs.

With the aid of the above explanation and the step response results (Figure 22) it is apparent that the stopping distances of the two-fluid brake system are longer because the response of the system is slower. The time required to reapply brake pressure from a pressure dump (after a skid) is longer. The increased time at lower pressure is lost braking performance which results in longer stopping distances.

Several tests were conducted to determine the effect of the two-fluid brake system configuration on stopping performance under variable runway conditions. The runway friction coefficient was varied during these tests to simulate wet

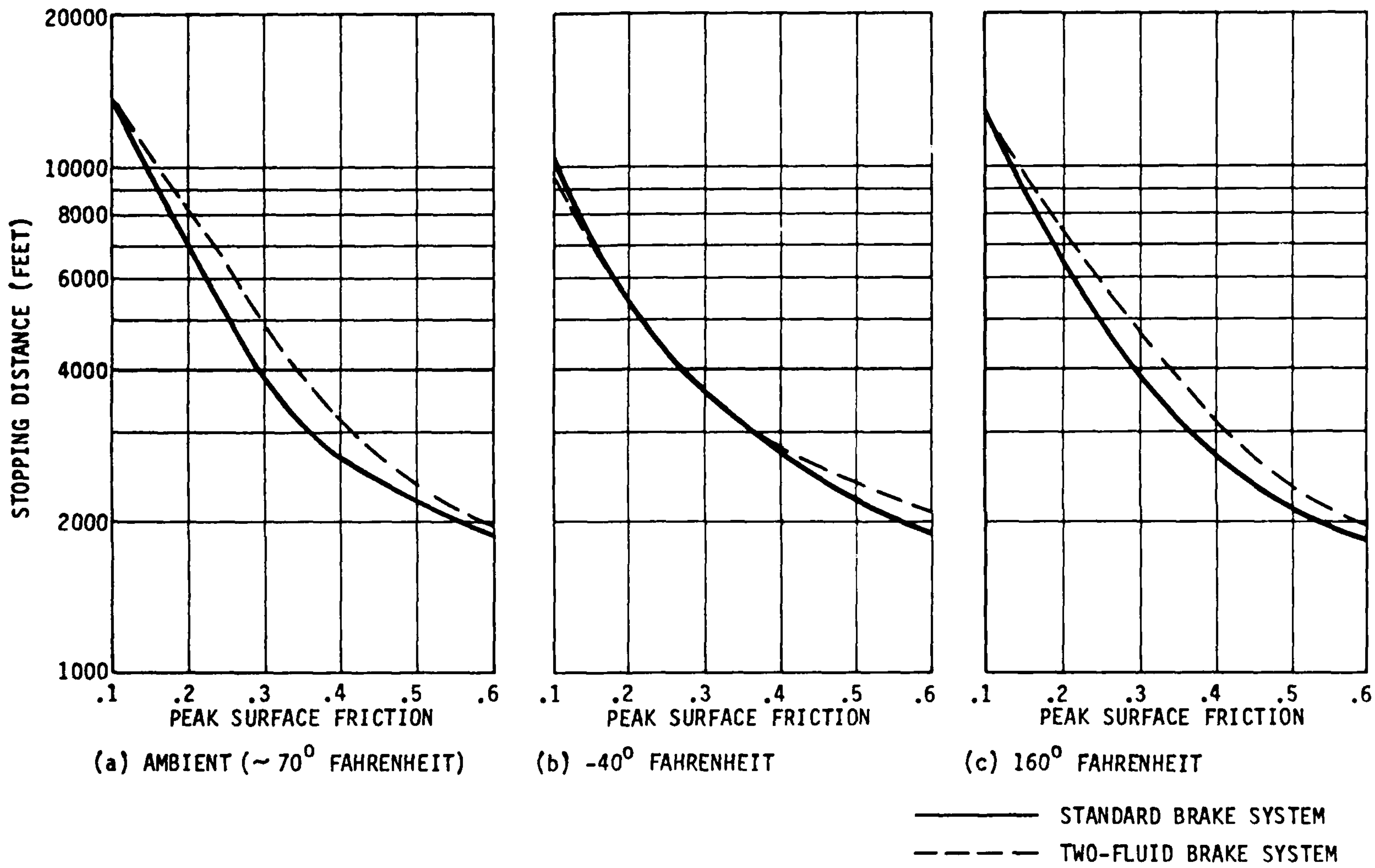


Figure 24 Effect of the Two-Fluid Brake System on Stopping Distance

TABLE 7 EFFECT OF THE TWO-FLUID BRAKE SYSTEM ON STOPPING PERFORMANCE

		STOPPING DISTANCE - FEET					
		AMBIENT		-40°F		+160°F	
TEST/DESCRIPTION	PEAK AVAILABLE RUNWAY FRICTION	STANDARD SYSTEM	TWO-FLUID SYSTEM	STANDARD SYSTEM	TWO-FLUID SYSTEM	STANDARD SYSTEM	TWO-FLUID SYSTEM
TEST 5 CONSTANT RUNWAY FRICTION	.6	1879	1935	1895	2066	1825	1901
	.5	2234	2353	2240	2382	2158	2296
	.4	2694	3120	2779	2837	2650	3197
	.3	3784	4816	3548	3604	3942	4668
	.2	7214	8508	5452	5422	6706	7520
	.1	13325	13220	10396	9799	12828	12430
TEST 6 WET RUNWAY	.1 to .5	4725	5543	4331	4380	4608	5028
	.1 to .35	5963	7098	5208	5255	5936	6453
TEST 7 STEP FRICTION	.1 to .5	3957	4521	6105	5917	3807	4133

TABLE 7 EFFECT OF THE TWO-FLUID BRAKE SYSTEM ON STOPPING PERFORMANCE

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		AMBIENT		-40°F		+160°F	
TEST/DESCRIPTION	PEAK AVAILABLE RUNWAY FRICTION	STANDARD SYSTEM	TWO-FLUID SYSTEM	STANDARD SYSTEM	TWO-FLUID SYSTEM	STANDARD SYSTEM	TWO-FLUID SYSTEM
TEST 5 CONSTANT RUNWAY FRICTION	.6	1879	1935	1895	2066	1825	1901
	.5	2234	2353	2240	2382	2158	2296
	.4	2694	3120	2779	2837	2650	3197
	.3	3784	4816	3548	3604	3942	4668
	.2	7214	8508	5452	5422	6706	7520
	.1	13325	13220	10396	9799	12828	12430
TEST 6 WET RUNWAY	.1 to .5	4725	5543	4331	4380	4608	5028
	.1 to .35	5963	7098	5208	5255	5936	6453
TEST 7 STEP FRICTION	.1 to .5	3957	4521	6105	5917	3807	4133

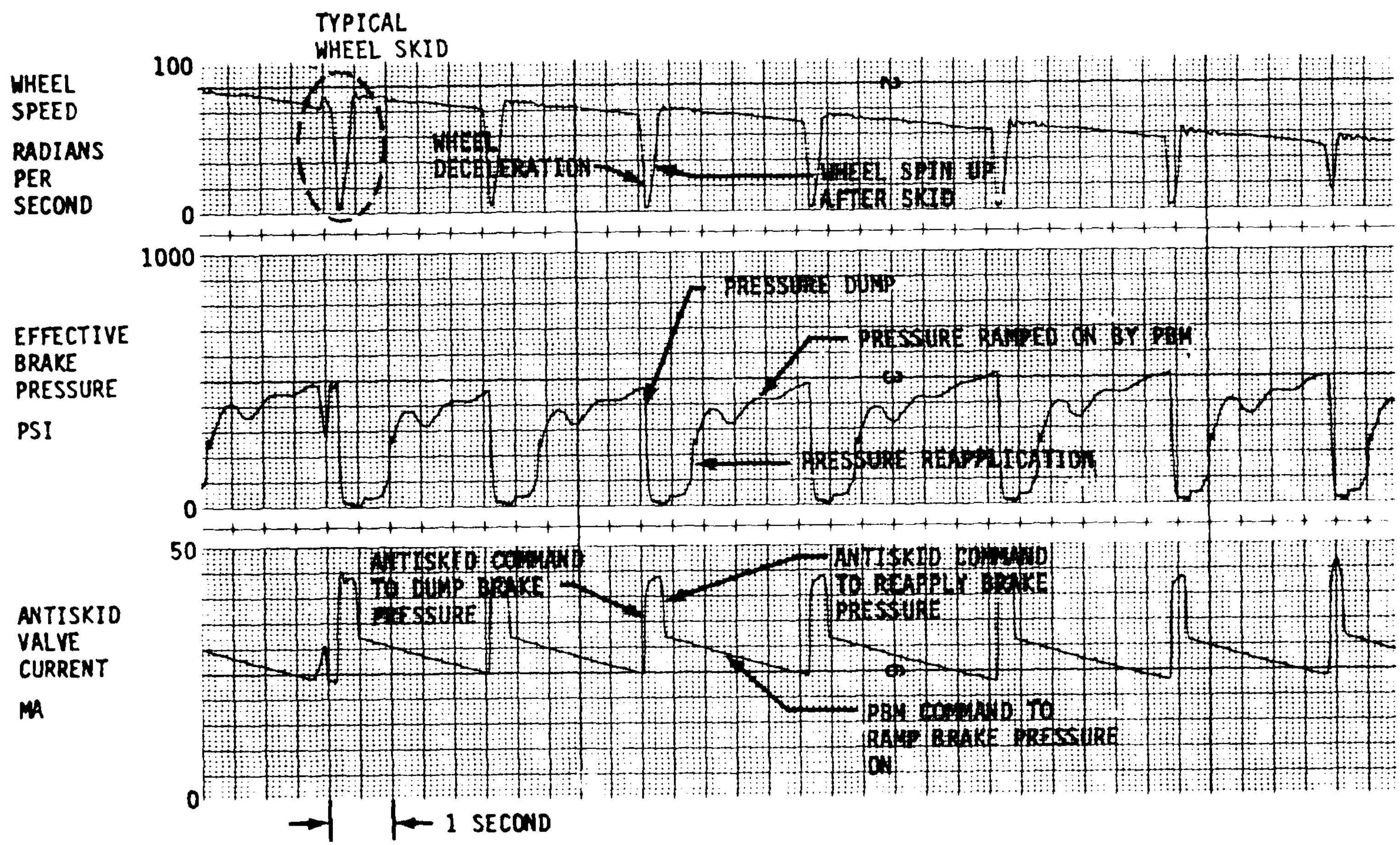

Figure 25 KC-135 Brake System Performance

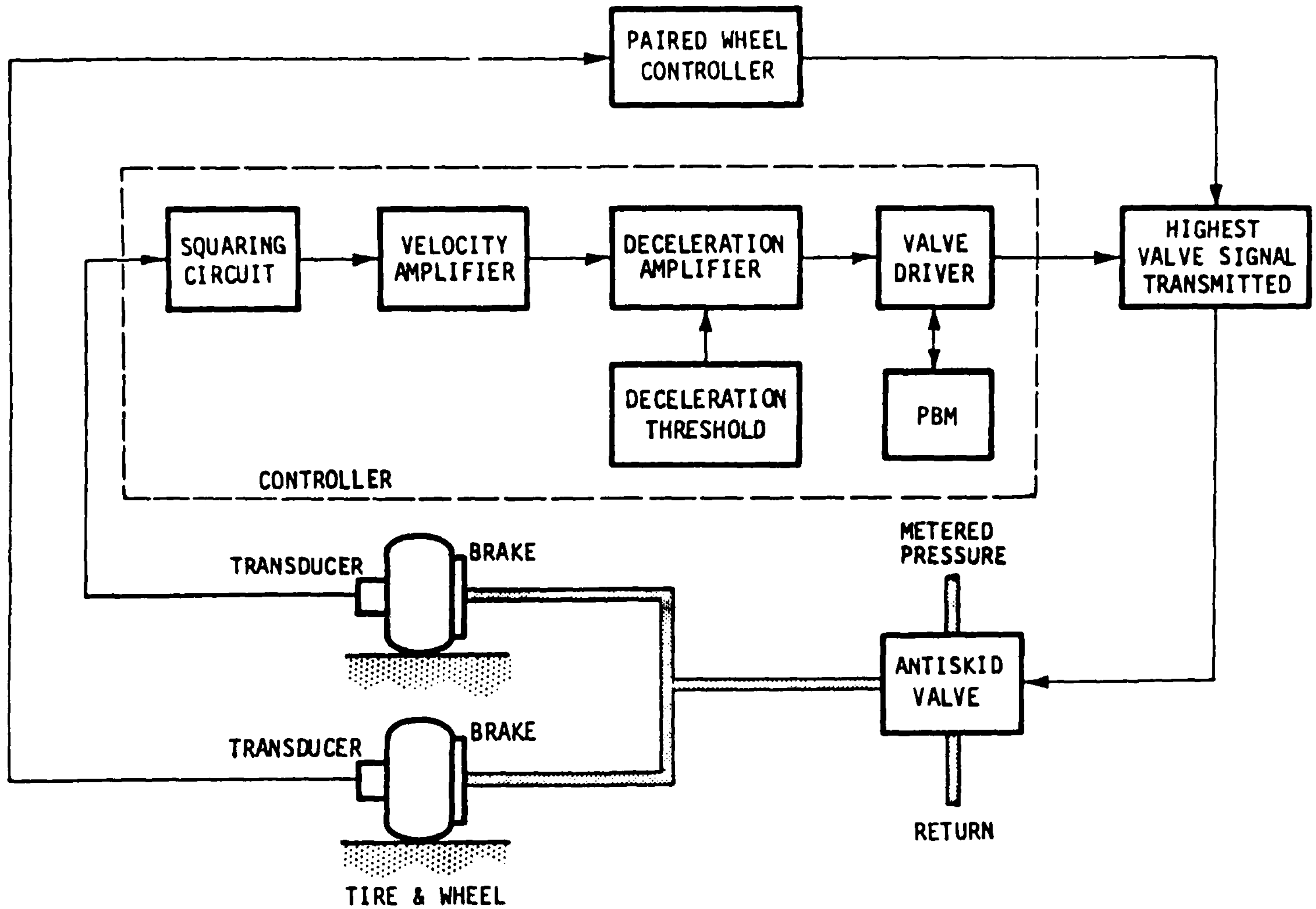

Figure 26 KC-135 Brake System Block Diagram

or icy runways (Test 6) and runways with wet or icy patches (Test 7). The results of the tests are given in Table 7. The stopping distances of the two-fluid system under variable conditions are longer than those of the standard system. The increase is again directly attributed to the reduction in brake hydraulic system dynamic response.

The percent increase in stopping distance of the two-fluid system over the baseline system is greatest at the values of runway friction of .3 and .4. This variation in stopping distance between the two-fluid system and the baseline is caused by a number of system effects that vary nonlinearly as a function of runway friction. Principal among these are the tire/runway friction characteristic as a function of tire slip (i.e., the mu-slip curve), the level of nominal operating brake pressure, and the brake hydraulic system dynamic response characteristic.

3.4.3 LANDING GEAR AND BRAKE CONTROL SYSTEM STABILITY

A series of tests were performed to determine the effect which the two-fluid brake system has upon the stability of the landing gear and the brake control system. Changes to the brake system can cause uncontrollable and divergent fore and aft oscillation of the landing gear (termed gear walk) which can result in a landing gear failure. Fore and aft motion of the gear also superimposes an oscillation on top of the wheel speed signal. When gear walk motion is severe (but not to the point of failure) the antiskid system can interpret the oscillations as repetitive skids leading to abnormal and unnecessary reductions in brake pressure.

The two-fluid brake system did not effect the landing gear or brake control system stability with normal strut damping included in the simulation. The fore and aft motion of the landing gear was not increased and no gear walk or abnormal brake pressure reductions occurred.

3.4.4 -65 DEGREES FAHRENHEIT TEST RESULTS

The low temperature system performance tests were attempted, as planned, at -65 degrees Fahrenheit. However, during the tests with the standard system,

and subsequently with the two-fluid system, a significant amount of fluid leakage from the brakes and antiskid valve occurred. This leakage allowed only partial completion of the -65 degree Fahrenheit low temperature tests and raised some question as to the validity of the results which were obtained. The results of these tests are documented in Appendix E. The tests were repeated at -40 degrees Fahrenheit without any leakage. Consequently, the -40 degree Fahrenheit test data were selected for the discussion of low temperature brake system performance presented in Section 3.4.1.

The effect of the leakage on the -65 degree Fahrenheit results and brake system performance is unknown. However, no difference in system operation was observed and the stopping distance data is qualitatively consistent with the -40 degrees Fahrenheit test data.

3.4.5 FLUID SAMPLES

CTFE hydraulic fluid samples were taken periodically during the two-fluid system performance tests. These fluid samples were submitted to the AFWAL/MLBT for analysis. A sample of unused CTFE was analyzed by the Boeing Company. The results of this analysis and a complete fluid sample history is given in Appendix H.

SECTION IV

RELIABILITY AND MAINTAINABILITY OF THE TWO-FLUID BRAKE SYSTEM

4.1 RELIABILITY AND MAINTAINABILITY STUDY

The two-fluid brake hydraulic system was evaluated to determine the potential reliability and maintainability of the new hardware compared with existing brake systems and components. During this analysis the following factors were considered.

(1) The effect of the two-fluid system and component hardware design upon the failure (reliability) rate, maintenance tasks and maintenance (repair) rate of the brake system.

(2) The effect of the two-fluid system on servicing and possible compromises in system reliability and maintainability due to servicing errors.

(3) The effect upon logistic support equipment.

(4) The effect of the fluid's physical, chemical and material properties upon component and system integrity and the effects of normal shop or service area contaminates upon the seal materials.

4.2 TWO-FLUID BRAKE SYSTEM FAILURE AND MAINTENANCE RATES

The relative reliability and maintainability of the two-fluid brake hydraulic system was estimated by comparing the system's predicted failure and maintenance rates with actual aircraft brake system data. (Normal system servicing and scheduled maintenance were not included in the reliability and maintainability study). The failure and maintenance rates of each component and the total system were estimated using data on existing devices and systems which are similar in function and construction. The estimates were based upon the assumption that the PNF seal service life and failure rate are the same as

the nitrile seals currently in service. KC-135A brake system Reliability and Maintainability Final Data (compiled per AFR66-1) obtained from the Air Force Logistics Center, WPAFB, Ohio, was used to evaluate the two-fluid brake system and hardware. This data covers the January 1978 to December 1978 time period and represents an inventory of 425 KC-135A aircraft with 146,435 flight hours during the 12 months time period.

Reliability and maintainability data for the standard KC-135A brake system (and each component) are given in Tables 8 and 9 respectively. These data are presented in terms of the number of maintenance tasks, elapsed maintenance times, maintenance man hours and failures per one thousand flight hours. A failure is a state or condition which occurs when a piece of hardware does not function as intended. A maintenance task is a shop or flight line action which is required to remove, diagnosis, repair or replace a component which has failed. The elapse maintenance time (EMT) is the clock time in hours required to perform maintenance task. The maintenance man hours (MMH) is the number of man hours required to perform a maintenance task (i.e., MMH = EMT x the number of men required to perform the task).

The KC-135A data along with engineering judgment were used to predict the reliability and maintainability of the two-fluid brake system (Tables 8 and 9). The effects of the modified deboost valve, the new replenishment reservoir, additional shuttle valves and restrictors and the removal of the hydraulic fuses are itemized in the tables. The reliability and maintainability of the hydraulic fuses, shuttle values and restrictors in the two-fluid system were determined by ratioing the standard KC-135A component data up or down to account for the number of units per aircraft. The reliability and maintainability data for the modified deboost valve and replenishment reservoir were estimated using the standard deboost valve data. The deboost valve is similar both in performance and construction to each of the new components. The deboost valve data were adjusted in each case to reflect the predicted increase in failures and maintenance due to additional seals, fittings, bolts and complexity.

TABLE 8 RELIABILITY DATA

SYSTEM COMPONENT	STANDARD KC-135 BRAKE SYSTEM QUANTITY PER A/P	STANDARD KC-135 BRAKE SYSTEM FAILURES PER 1000 FH	KC-135 TWO-FLUID FIREPROOF BRAKE SYSTEM QUANTITY PER A/P	KC-135 TWO-FLUID FIREPROOF BRAKE SYSTEM FAILURES PER 1000 FH
BRAKE SYSTEM (TOTAL)		63.202		64.347
BRAKE ASSEMBLY	8	55.349	U	U
HOSE ASSEMBLY	4	1.694	U	U
VANE, DUAL CONTROL	2	.055	U	U
BRAKE METERING VALVE	4	.539	U	U
HYDRAULIC FUSE	8	.232	0	0.0
SHUTTLE VALVE	4	.020	U	U
DEBOOST VALVE	4	1.147	0	0.0
PRESSURE RELIEF VALVE	1	.212	U	U
RESERVE BRAKE ACCUMULATOR	1	1.550	U	U
RESTRICTOR	*	*	U	U
CHECK VALVE	1	.055	U	U
OTHER		2.349	U	U
NEW TWO-FLUID SYSTEM COMPONENTS				
DEBOOST VALVE WITH REPLENISHMENT VALVE			4	1.367
REPLENISHMENT RESERVOIR			4	1.147
RESTRICTOR			4	0.0
SHUTTLE VALVE			2	.010

* - NO DATA OR NO FAILURES
U - UNCHANGED
A/P - AIRPLANE
FH - FLIGHT HOUR

TABLE 9 MAINTAINABILITY DATA

SYSTEM COMPONENTS	STANDARD KC-135 BRAKE SYSTEM QUANTITY PER A/P	MAINTEN. TASKS PER 1000 FH	EMT PER 1000 FH	MMH PER 1000 FH	KC-135 TWO-FLUID FIREPROOF BRAKE SYSTEM QUANTITY PER A/P	MAINTEN. TASKS PER 1000 FH	EMT PER 1000 FH	MMT PER 1000 FH
BRAKE SYSTEM (TOTAL)		146.543	134.894	259.706		146.595	138.578	267.111
BRAKE ASSEMBLY	8	117.807	106.143	202.360	U	U	U	U
HOSE ASSEMBLY	4	3.203	2.934	5.544	U	U	U	U
VALVE DUAL CONTROL	2	.116	.105	.180	U	U	U	U
BRAKE METERING VALVE	4	2.267	1.971	3.884	U	U	U	U
HYDRAULIC FUSE	8	5.716	1.959	3.781	0	0.0	0.0	0.0
SHUTTLE VALVE	4	.102	.080	.148	U	U	U	U
DEBOOST VALVE	4	4.924	4.811	9.480	0	0.0	0.0	0.0
PRESSURE RELIEF VALVE	1	.820	.873	1.644	U	U	U	U
RESERVE BRAKE ACCUMULATOR	1	3.852	7.020	14.191	U	U	U	U
RESTRICTOR	*	*	*	*	U	U	U	U
CHECK VALVE	1	.198	.198	.408	U	U	U	U
OTHER		7.539	8.800	18.086	U	U	U	U
NEW TWO-FLUID SYSTEM COMPONENTS								
DEBOOST VALVE WITH REPLENISHMENT VALVE					4	5.716	5.603	11.112
REPLENISHMENT RESERVOIR					4	4.924	4.811	9.480
RESTRICTOR					4	0.0	0.0	0.0
SHUTTLE VALVE					2	.051	.040	.074

* - NO DATA OR NO MAINTENANCE ACTION
U - UNCHANGED
A/P - AIRPLANE

EMT - ELAPSE MAINTENANCE TIME
MMH - MAINTENANCE MAN HOURS
FH - FLIGHT HOUR

Total brake system figures indicate that the two-fluid brake hydraulic system configuration will have very little effect on system reliability and maintainability. The predicted number of failures, elapse maintenance time and maintenance man hours per one thousand flight hours are each increased approximately three percent due primarily to the addition of the replenishment reservoir. The number of maintenance tasks per one thousand flight hours is nearly unchanged due to balance between the removal of the hydraulic fuses and the addition of the replenishment reservoir and the modified deboost valve.

4.3 SERVICING

The two-fluid brake system has been designed to minimize the impact on brake system servicing. Servicing of the two-fluid brake system is required only when the CTFE fluid level in the replacement fluid reservoir is low. The replenishment reservoir has been sized to accommodate volumetric changes due to normal brake wear, temperature changes and minor (acceptable) fluid leakage. Consequently, servicing (i.e., fluid replacement and bleeding) of the two-fluid brake system will only be required when a worn brake is replaced (servicing only the affected portion of the brake system), a seal fails or a line bursts. The standard brake system also requires servicing in these cases.

Servicing of the two-fluid brake system is slightly more complex and time consuming than the standard system. After servicing the primary (MIL-H-5606) fluid system additional steps are required to service the CTFE portion of the brake system (see Section 2.4.4 of the Interim Technical Report, Appendix A). The additional time required for servicing the CTFE portion of the brake system is estimated to be forty minutes (ten minutes per replenishment reservoir). Servicing of the normal brake system is estimated to require approximately two to three hours.

The incorporation of the two-fluid brake hydraulic system presents a new servicing hazard not present in the standard single-fluid system. This problem involves the inadvertent mixing of the CTFE fluid with the MIL-H-5606 fluid in the brake system. Depending upon the particular CTFE fluid used, the

effect on the system can be detrimental, such as forming a precipitate. However, the CTFE fluid used for this study (Halocarbon AO-2) can be mixed with MIL-H-5606 fluid without the formation of a precipitate. The possibility of fluid mixing can be reduced (but not totally eliminated) by using a different type of fitting on the servicing ports of the CTFE portion of the hydraulic system and the CTFE fluid service unit so that the service unit cannot be connected to MIL-H-5606 fluid service ports.

4.4 SUPPORT EQUIPMENT

The two-fluid brake hydraulic system will require an additional ground cart for servicing the CTFE fluid. Due to the small quantity of CTFE fluid in the brake system the cart may be a small portable wheeled cart containing a fluid reservoir, hand-driven hydraulic pump and filter.

4.5 MATERIALS COMPATIBILITY

The CTFE fluid has no effect on the physical, chemical or material properties of the metals (aluminums and steels), paints, lubricants and greases normally found on the aircraft. The CTFE fluid may, however, have some affect on rubber or elastomer materials such as seals, hoses and clamps and would have to be investigated. Such items exposed to the CTFE may require redesign or protective coverings.

The PNF seals used in the CTFE portion of the two-fluid brake hydraulic system are compatible with the CTFE, MIL-H-5606, and MIL-H-83282 hydraulic fluids. In addition the seals are resistant to unleaded and high peroxide content gasolines, Arctic diesel fuel, conventional and synthetic jet fuels and dieter synthetic and E.P. gear lubricants (Reference 2) which may be found in shop or rebuild areas.

SECTION V

CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

(1) Based on the analyses and laboratory tests performed in this program, the concept of a fireproof two-fluid brake hydraulic system has been shown to be feasible. A two-fluid brake system was built and successfully tested in a laboratory environment.

(2) The basic operation and control characteristics of the brake system are not affected by the two-fluid configuration. Physical modifications to hardware and the use of PNF seals have virtually no affect on the brake system performance.

(3) The dynamic response of the brake hydraulic system and the stopping performance of the airplane are affected by the used of CTFE fluid. The increased density of CTFE fluid (CTFE is 2.11 times as dense as MIL-H-5606 fluid) causes the hydraulic system to respond slower resulting in longer aircraft stopping distances. The largest increase in stopping distance, (determined in laboratory tests at a runway friction coefficient of .3 and at ambient temperature) was approximately 27 percent.

(4) The performance lost by changing to the CTFE fluid can be regained by increasing the area of hydraulic lines in the CTFE portion of the brake system by a factor of 2.11 and retuning the antiskid control box.

(5) The CTFE and MIL-H-5606 hydraulic fluids can be effectively isolated by the use of a floating piston type device. The feasibility of an isolation unit was demonstrated during laboratory testing.

(6) A two-fluid brake hydraulic system configured for the KC-135 aircraft (with appropriate modifications to hydraulic lines, etc., to maintain the current stopping performance) will increase the airplane weight approximately 64 pounds.

(7) The two-fluid brake hydraulic system has a minimal effect on the reliability (failure rate) and maintainability of the brake system. The two-fluid system configured for the KC-135 was estimated to cause approximately three percent increases in the failure and maintenance rates of the brake system.

5.2 RECOMMENDATIONS

The feasibility of the fireproof two-fluid brake hydraulic system has been demonstrated. It is recommended that a program for the continued development of this technology include the following design and testing efforts.

(1) Perform a design study to develop the optimum two-fluid brake system design considering all performance factors identified in this report. This would include determining required system modifications to achieve stopping performance comparable to the baseline aircraft.

(2) Perform laboratory tests (using brake system hardware) to tune the antiskid controller and verify brake system stopping performance. In addition, laboratory tests should be performed to demonstrate the replenishing system and servicing procedures.

(3) Perform aircraft ground roll tests to verify and demonstrate the two-fluid brake system performance exposed to a variety of field conditions.

APPENDIX A

INTERIM TECHNICAL REPORT

The Interim Technical Report which was submitted to and approved by the Air Force is reprinted in the following pages.

The Interim Technical Report contains a complete description of the two-fluid brake hydraulic system design, the computer analysis of the two-fluid system, recommended component modifications for testing, and a synopsis of the component and system test plans.

FIREPROOF BRAKE HYDRAULIC SYSTEM

INTERIM TECHNICAL REPORT

(REVISED)

CONTRACT F33615-80-C-2026
PROJECT 3145
CDRL SEQUENCE NO. 6

NOVEMBER 1980

REVISED JANUARY 1981

SUBMITTED FOR APPROVAL OF:

AIR FORCE AERO PROPULSION LABORATORY
ATTN: BRUCE CAMPBELL, AFWAL/POOS-1

PRINCIPAL INVESTIGATOR Steven M. Warren PROGRAM MANAGER J R Kilner
S. M. WARREN J. R. KILNER

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SECTION I

INTRODUCTION

This interim Technical Report is submitted to the Air Force in compliance with CDRL Sequence Number 6 and meets the requirements specified in Section C, Paragraphs 4.6 and 5.2 of Contract F33615-80-C-2026.

1.1 PROGRAM OBJECTIVE AND STRUCTURE

The objective of the Fireproof Brake Hydraulic System study is to determine the feasibility of two-fluid brake hydraulic system which uses a nonflammable chlorotrifluoroethylene (CTFE) base hydraulic fluid in the high fire potential area (i.e., in the immediate area of the brake). The program has been divided into six tasks (Table 1.1) concluding in a laboratory evaluation of the operation and performance of an actual two-fluid brake hydraulic system.

1.2 SCOPE OF INTERIM TECHNICAL REPORT

This Interim Technical Report presents the work accomplished during the Component Assessment and Redesign Task (Task 2) and the Component and System Test Plan Task (Task 3). This work was performed in preparation for the laboratory evaluation of two-fluid brake hydraulic system and included

(1) The configuration of a two-fluid brake hydraulic system,

(2) The analytical assessment of system dynamic response,

(3) The design modifications to an existing laboratory brake hydraulic system required for testing and evaluation of the two-fluid system concept, and

(4) The development of a component and system performance test plan.

TABLE 1.1 FIREPROOF BRAKE HYDRAULIC SYSTEM TASKS

TASK 1	ACQUISTION OF BRAKE COMPONENTS
TASK 2	COMPONENT ASSESSMENT AND REDESIGN FOR THE TWO-FLUID SYSTEM
TASK 3	COMPONENT AND SYSTEM TEST PLAN
TASK 4	COMPONENT MODIFICATIONS AND TEST
TASK 5	COMPONENT INSTALLATION AND SYSTEM TEST
TASK 6	RELIABILITY/MAINTAINABILITY STUDY

1.3 STUDY AIRCRAFT

The initial step in this program is to select a brake hydraulic system representative of a modern-day cargo aircraft which can be modified and tested to evaluate the two-fluid brake hydraulic system concept. The KC-135 brake system has been selected for modification and testing. This selection was performed during the proposal period and was included in the Boeing Proposal as the recommended study system.

The KC-135 was selected due to the existence of a deboost valve which can be easily modified to act as a fluid isolator in the two-fluid system.

1.4 CONTENTS OF REPORT

This report contains:

- A description of a preliminary two-fluid brake hydraulic system design in Section II.

- An analytical assessment of the two-fluid configuration and a fluid property sensitivity study (supporting the selection of a specific chlorotrifluoroethylene base hydraulic fluid), in Section III.

- The recommended modifications necessary to convert the KC-135 brake hydraulic system to a two-fluid configuration for laboratory performance testing in Section IV.

- A description of the component Performance and System Performance Test Plan in Section V.

SECTION II

TWO-FLUID BRAKE HYDRAULIC SYSTEM DESIGN

2.1 SYSTEM DESIGN OBJECTIVE

The objective of the system design effort (Task 2) was to define a realistic two-fluid brake hydraulic system configuration which can be used to determine and evaluate the effects which the two-fluid concept has upon brake system operation and performance. Specifically, the task involved redesigning the existing KC-135 brake hydraulic system to incorporate the two-fluid concept.

The system redesign effort and the selection of a two-fluid system configuration was approached with the following goals in mind:

(1) To minimize the changes to the existing brake hydraulic system,
(2) To keep system fabrication and material costs low, and
(3) To design system changes that can be easily retrofitted on existing aircraft.

2.2 SYSTEM DESIGN REQUIREMENTS

The two-fluid brake hydraulic system was designed to meet the following component and system requirements:

(1) To provide positive and reliable separation of the two hydraulic fluids (MIL-H-5606 from the AO-8* fluid).
(2) To operate in a temperature range of -65 degrees F. to 160 degrees F. (the range of brake system operation).
(3) To provide an AO-8 fluid replenishment system which accounts for volumetric changes due to temperature, brake wear and normal fluid loss (leakage).
(4) To meet present aircraft reliability and safety standards.

*Halocarbon AO-8 or Halocarbon AO-2 (both CTFE base hydraulic fluids) are being considered for use in the two-fluid brake system.

2.3 EXISTING KC-135 BRAKE SYSTEM

The KC-135 brake system is shown in Figure 2.1. The KC-135 employs a truck type main landing gear with paired wheel control. That is, the brake pressure associated with each forward and aft wheel pair on one side of the truck is controlled by a single antiskid valve and control system. The hydraulic system associated with a single-tandem-wheel pair is shown in Figure 2.2.

A principal feature of the KC-135 brake hydraulic system is the deboost valve. The deboost valve is a pressure reducing device which transforms 3000 psi pressure signals from the antiskid valve to 965 psi signals at the brake assembly. The deboost valve has a replenishment valve (flow path) which permits fluid to flow from high pressure to low pressure when makeup fluid is required in the normally isolated low pressure volume. This deboost valve can be easily modified to act as a fluid isolator (by plugging the replenishment valve flow path) between the AO-8 fluid in the brake assembly and the MIL-H-5606 fluid in the aircraft hydraulic system.

2.4 TWO-FLUID BRAKE HYDRAULIC SYSTEM

Several preliminary KC-135 two-fluid brake hydraulic system configurations were designed. These were evaluated based on factors including maintainability, reliability, retrofit cost and risk associated with each design. The best preliminary design was selected for analytical assessment and detailed design.

The analytical assessment was performed to determine (1) the effect which the proposed two-fluid system has upon the dynamic response of the brake hydraulic system, and (2) the modifications to the two-fluid system configuration which are required to achieve a dynamic response characteristic comparable to the KC-135 brake hydraulic system. The results of the analytical study are discussed in Section III. The final KC-135 two-fluid brake hydraulic system configuration is discussed below.

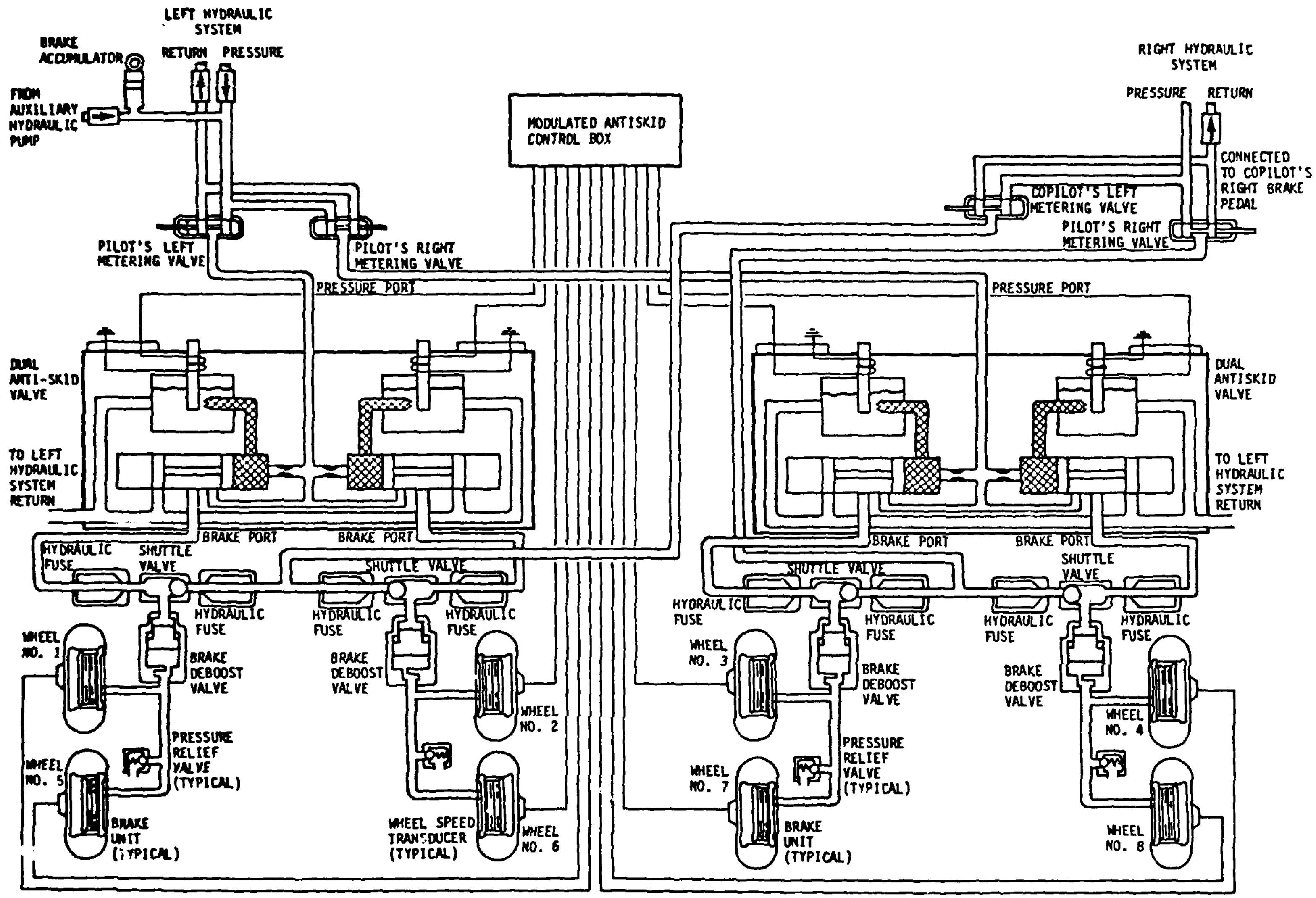

Figure 2.1 KC-135 Brake System Schematic

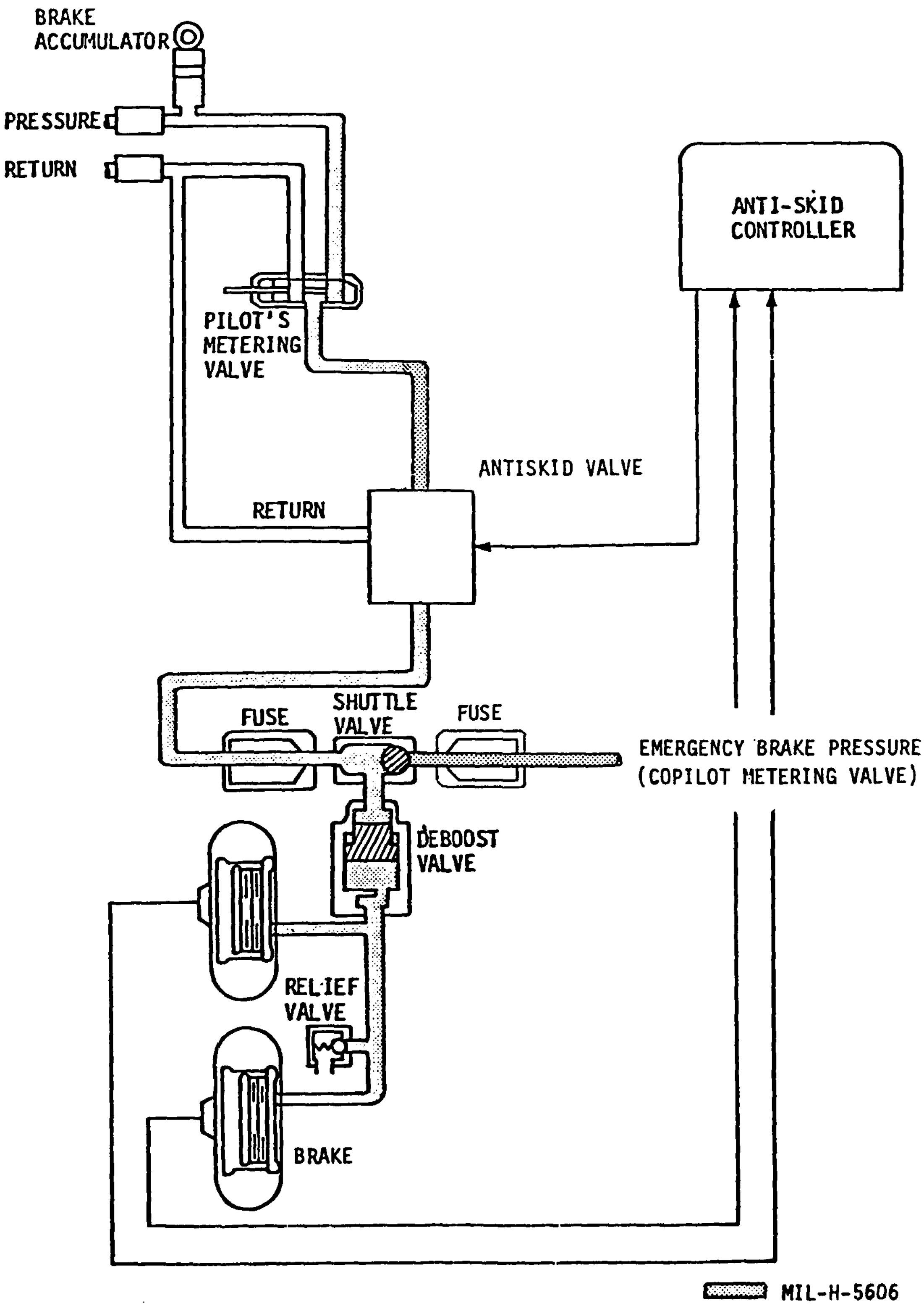

Figure 2.2 KC-135 Brake System, Single-Tandem-Wheel Pair

2.4.1 BASIC TWO-FLUID SYSTEM CONFIGURATION

The two-fluid brake hydraulic system associated with a single-tandem-wheel pair is shown in Figure 2.3 while a schematic of the entire KC-135 two-fluid brake system is shown in Figure 2.4.

The major modifications necessary to convert the KC-135 brake hydraulic system to the two-fluid configuration are:

(1) The elimination of the original fluid flow path through the deboost valve (i.e., fluid replenishment valve and pin; see Figure 2.5).

(2) The addition of a separate AO-8 fluid replenishment system (i.e., replenishment valve in the new end cap and a replenishment reservoir as shown in Figure 2.6).

(3) The use of PNF O-ring seals in area exposed to the AO-8 fluid.

Each tandem-wheel pair has its own AO-8 fluid replenishment system (Figure 2.4). The replenishment system is pressurized normally by pilot metered pressure; however, in the event of a failure the reservoir is pressurized by the copilot metered pressure. Since the fluid flow path from the high pressure side to the low pressure side of the deboost valve has been plugged, the need for the quantity measuring fuses (in the two-fluid system) upstream of the deboost valve is eliminated.

This configuration has two distinct advantages: (1) the brake hydraulic system is virtually unchanged and (2) it functions exactly the same as the original KC-135 system. No modifications have been made which affect or change the dynamic operation of the deboost valve or brake system. During normal brake system operation, the original KC-135 brake system and the modified two-fluid KC-135 brake hydraulic system are identical. The differences which exist between the configurations involve only the replenishment system (i.e., the original replenishment valve has been moved from the deboost valve piston to the end cap). Since the replenishment valve is closed (blocking the

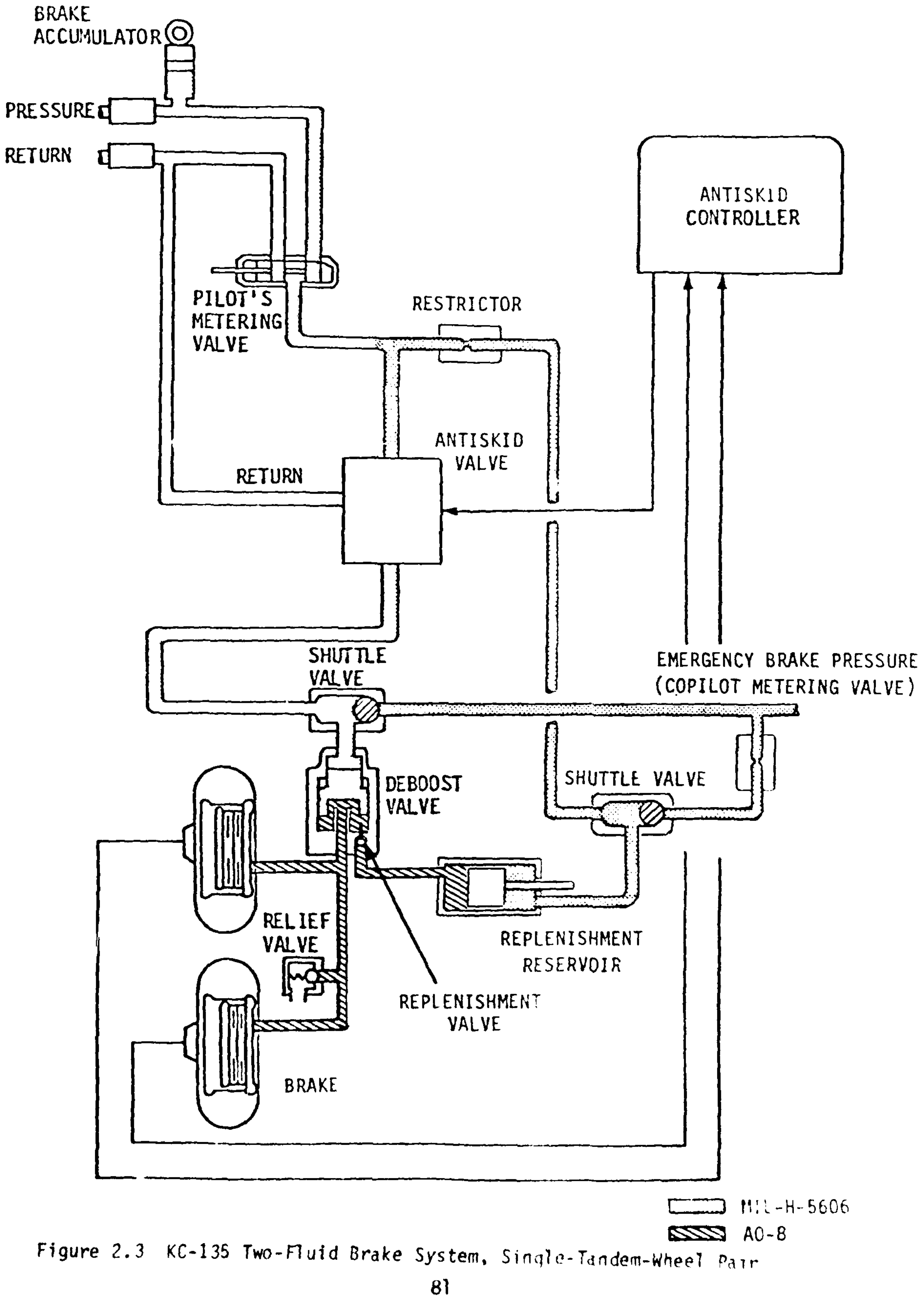

Figure 2.3 KC-135 Two-Fluid Brake System, Single-Tandem-Wheel Pair

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FIREPROOF BRAKE HYDRAULIC SYSTEM.(U)
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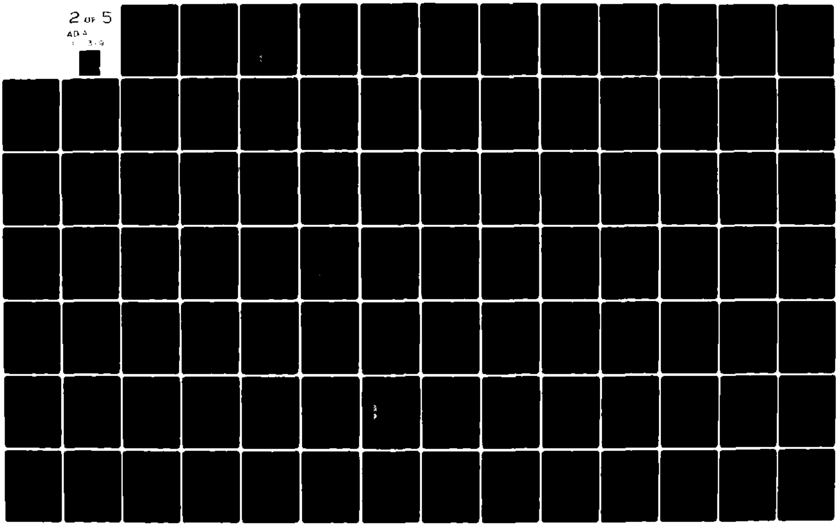

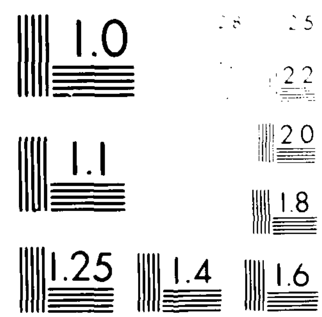
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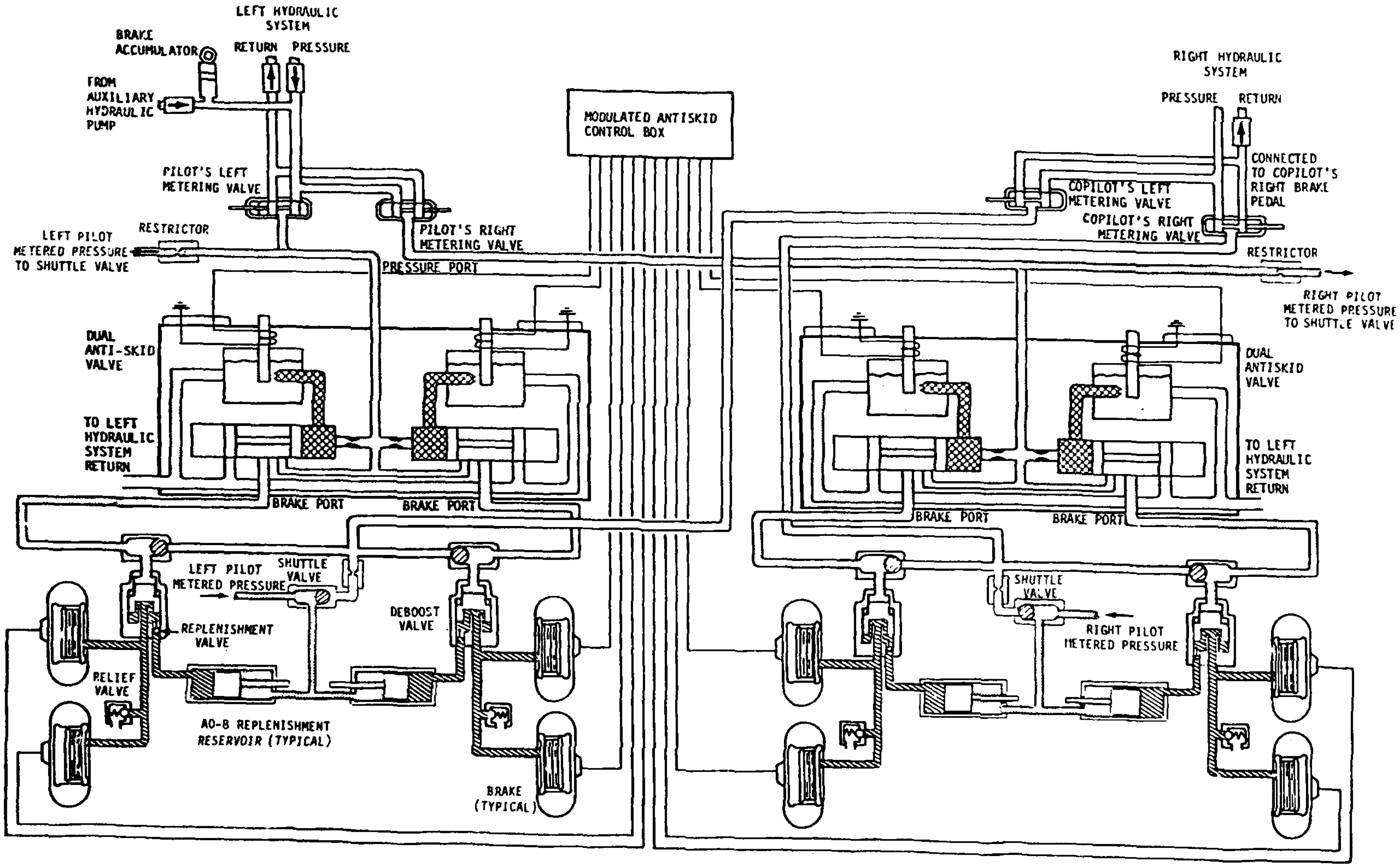


Figure 2.4 KC-135 Two-Fluid Brake System Schematic

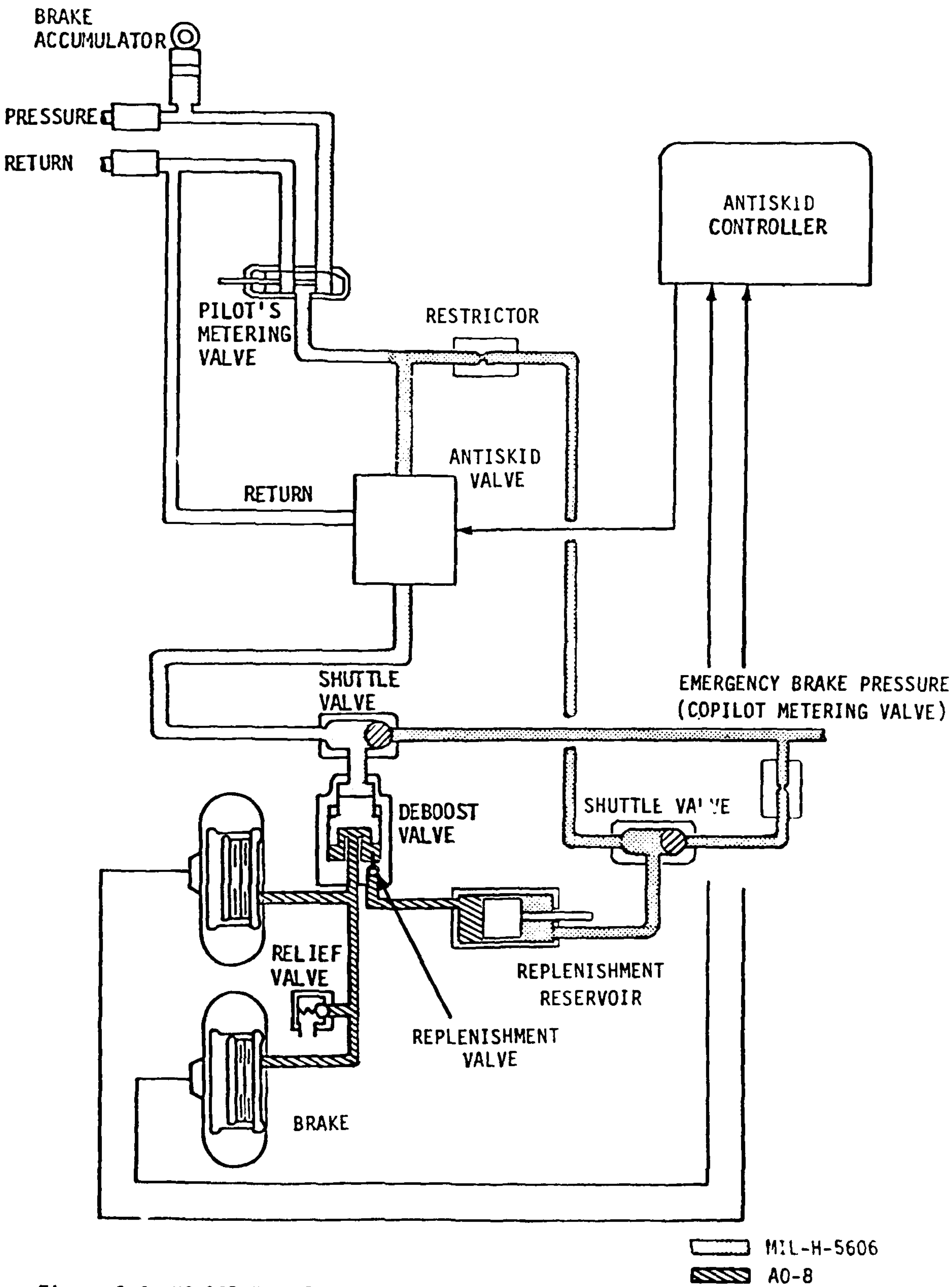


Figure 2.3 KC-135 Two-Fluid Brake System, Single-Tandem-Wheel Pair

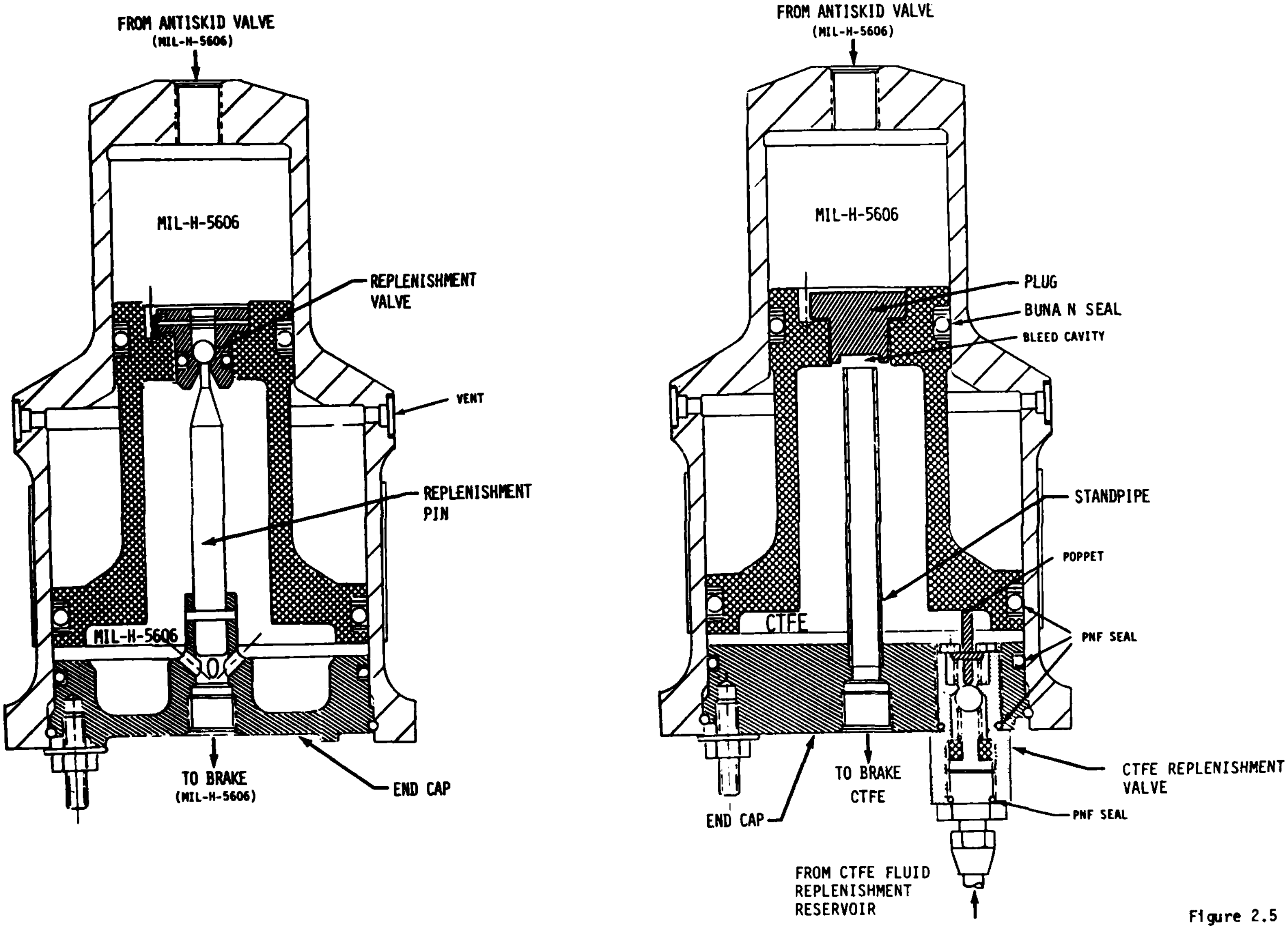


Figure 2.5

Deboost Valve Modifications

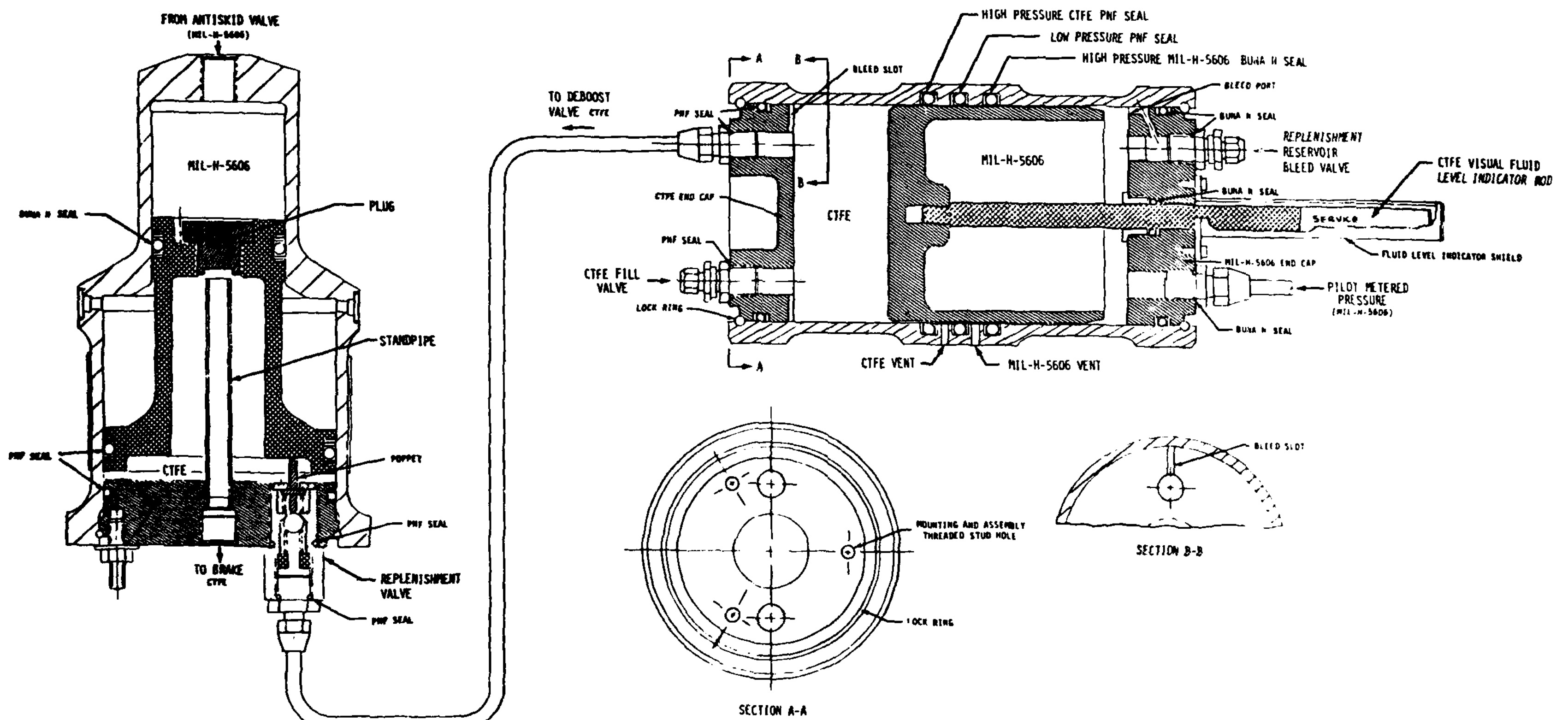


Figure 2.6 Deboost Valve and AO-8 Replenishment System

replenishment path in both systems) during normal braking activity, the configuration of the systems are identical. Thus, the brake and deboost valve modifications do not affect the normal operation of the brake system or the stopping performance of the aircraft. Similarly other brake system operating modes such as parking, refused takeoff, manual braking and emergency braking are not affected by the hardware modifications.

2.4.2 DEBOOST VALVE MODIFICATIONS

The modifications which convert the KC-135 deboost valve (Federal Stock Number 1650-00-570-8397) to a fluid isolator for the two-fluid system and the added replenishment system are shown in Figures 2.3 thru 2.6. These modifications are:

1) The original replenishment valve has been removed and a solid plug installed in its place to eliminate the original fluid interchange path (Figure 2.5).

2) The original end cap including the replenishment pin has been discarded (Figure 2.5).

3) A new end cap assembly has been manufactured. The end cap includes a bleed/output standpipe and a replenishment valve assembly (Figures 2.5 and 2.6).

4) The end cap and low pressure deboost valve piston seals have been changed to PNF materials (Figure 2.5).

5) An AO-8 fluid replenishment system including a fluid reservoir, refill valve for servicing and a fluid level indicator has been added (Figure 2.6).

6) Provisions for bleeding the isolated AO-8 fluid volume have been provided (Figure 2.6).

Although the deboost valve has been modified and an AO-8 fluid replenishment reservoir included, the system is virtually unchanged and functions exactly the same as the original KC-135 system. The replenishment valve has simply been moved from the top of the piston to the end cap to accommodate refilling from the replenishment fluid reservoir. The reservoir is pressurized by pilot metered pressure (Figure 2.3). Replenishment, as in the original system, occurs only when the piston is within 0.125 inch of the bottomed position at the low pressure end of the deboost valve. Replenishment fluid enters the low pressure end at the same pressure (not considering antiskid activity) as in the original system (i.e., approximately 3000 psi). In addition, during braking the deboost piston rides or functions at the same level (near the replenishment activation level) and with the same stroke as the unmodified deboost valve.

PLUG

The original replenishment valve in the piston has been removed and replaced with a plug to eliminate the fluid flow path from the high pressure (MIL-H-5606) to low pressure (AO-8) volume. The plug is designed as a shrink fit (0.002 to 0.003 interference) in the original valve piston. The threaded area in the deboost piston used to retain the original replenishment valve must be machined smooth to accommodate the shrink fit plug. A cavity is machined on the low pressure side of the plug to accommodate filling and bleeding the low pressure side of the two-fluid brake hydraulic system.

END CAP AND STANDPIPE

The original end cap and replenishment pin have been removed and replaced with a new end cap which contains the new replenishment valve assembly and the bleed/output standpipe. The end cap and standpipe is a welded assembly designed to minimize manufacturing costs. The standpipe is a section of standard 3/8 inch O.D. tubing which is machined to length, inserted in the end cap and then welded in place. The function of the standpipe is to bleed air from the low pressure (AO-8 fluid) side of the deboost valve. When the

deboost valve piston is bottomed against the end cap, the standpipe enters the plug cavity with a clearance of approximate .063 inch at the top. The replenishment valve assembly is designed to screw into the new end cap.

REPLENISHMENT SYSTEM

The replenishment system is composed of a replenishment valve and a pressurized fluid reservoir. The replenishment valve is a poppet actuated spring loaded ball type check valve which screws into the new end cap. When the poppet is depressed by the deboost valve piston, the spring loaded ball valve opens allowing fluid to flow from the reservoir to the deboost valve. This type of poppet actuated ball valve is commonly used in hydraulic systems. However, the valve is unique and must be manufactured especially for this application.

The replenishment reservoir (Figure 2.6) is a fluid volume (AO-8 fluid) which is pressurized by pilot metered pressure (MIL-H-5606 fluid). The reservor consists of a cylinder, two end caps, a piston, and an indicator rod and shield. The two-fluid isolator concept has been employed in the design to eliminate the need for a high pressure air charged AO-8 accumulator (for replenishment) which would cause the brake to lock if leakage occurred through the replenishment valve. However, when the two-fluid replenishment reservoir concept is used the reservoir is pressurized and replenishment occurs only when braking is commanded. Thus, when braking is not commanded (as in flight) the entire brake system (pilot metered pressure, the reservoir and brake) is at return pressure and no brake pressure build up can occur (due to replenishment valve leakage).

An AO-8 fluid level indicator rod is attached to the piston separating the AO-8 and MIL-H-5606 fluids. As the AO-8 reservoir fluid volume decreases the rod moves, exposing a red portion of the rod and the word SERVICE in the shield view port.

Two PNF O-rings and a Buna N nitrile O-ring are used to seal the piston and prevent leakage. Should leakage occur, two annular grooves and vent holes (one on either side of the low pressure PNF O-ring) are included to drain the fluid.

Each reservoir end cap contains two ports. The MIL-H-5606 cap contains a MIL-H-5606 supply port and a bleed port, while the AO-8 cap has a fill port and an AO-8 output port. Two slots are machined in the AO-8 end cap for bleeding (the reservoir is filled and bled with the piston against the AO-8 end cap). Similarily a drilled passage is included in the MIL-H-5606 end cap for bleeding.

The volume of AO-8 fluid contained in the reservoir has been sized to account for volumetric changes due to temperature, brake wear and normal fluid loss (leakage). The calculations and numbers supporting the selection of a 30 cubic inch reservoir volume per two-wheel set are given in Table 2.1. The volumes associated with brake wear and thermal expansion are well defined and can be accurately calculated. However, the leakage volume is an estimate largely dependent upon engineering experience, servicing practices and system design.

PILOT METERED PRESSURE RESTRICTOR

Pilot metered pressure is supplied to the replenishment reservoir through a restrictor (Figure 2.3). The restrictor is included to eliminate the dynamic effect which the additional MIL-H-5606 fluid volume in the replenishment reservoir has upon the response and performance of the brake hydraulic system.

2.4.3 BRAKE MODIFICATIONS

The KC-135 five rotor brake assembly (Federal Stock Number 1630-058-5242) requires no design modification for use in the two-fluid brake hydraulic system. However, since AO-8 hydraulic fluid will be used in the brake, the brake must be assembled with compatible seals. The brake seals which must be changed are shown in Figure 2.7.

2.4.4 TWO-FLUID BRAKE HYDRAULIC SYSTEM FILL AND BLEED PROCEDURE

Filling and bleeding the brake system is accomplished by ground servicing of the MIL-H-5606 portion of the system and then the AO-8 portion. Servicing the

TABLE 2.1 AO-8 REPLENISHMENT RESERVOIR CAPACITY

BRAKE SYSTEM DIMENSION AND VOLUMES

BRAKE PISTON AREA (1.375 INCH I.D. PISTON)	1.48 IN**2
PISTON AREA PER BRAKE ASSEMBLY (8 PISTONS)	11.87 IN**2
PISTON DISPLACEMENT	
NEW BRAKE DISPLACEMENT	0.25 IN
WORN BRAKE DISPLACEMENT	0.74 IN
BRAKE HOUSING VOLUME (DRILL PASSAGEWAYS)	2.36 IN**2
BRAKE LINE AND HOSE VOLUME (DEBOOST VALVE TO BRAKE,	44.73 IN**3
DEBOOST VALVE FLUID VOLUME (AO-8 FLUID)	15.04 IN**3
MAXIMUM BRAKE VOLUME (2 BRAKE ASSEMBLIES)	22.29 IN**3

TOTAL VOLUME FOR BRAKE WEAR (2 BRAKE ASSEMBLIES), VW

VW = 2 (11.87) (.74-.25) = 11.63 IN**3

TOTAL VOLUME FOR THERMAL CONTRACTION, VT

AO-8 COEFFICIENT OF THERMAL EXPANSION	0.0005 1/°F
MAXIMUM SERVICE TEMPERATURE (ASSUMED)	100 °F
MINIMUM OPERATING TEMPERATURE	-65 °F

VT = (15.04 + 44.73 + 22.29) X (100 - (-65))X (.0005) = 6.8 IN**3

TOTAL VOLUME FOR LEAKAGE (ESTIMATED), VL

VL = 10 IN**3

REPLENISHMENT RESERVOIR CAPACITY, VR

VR = VW + VT + VL = 28.43 IN**3

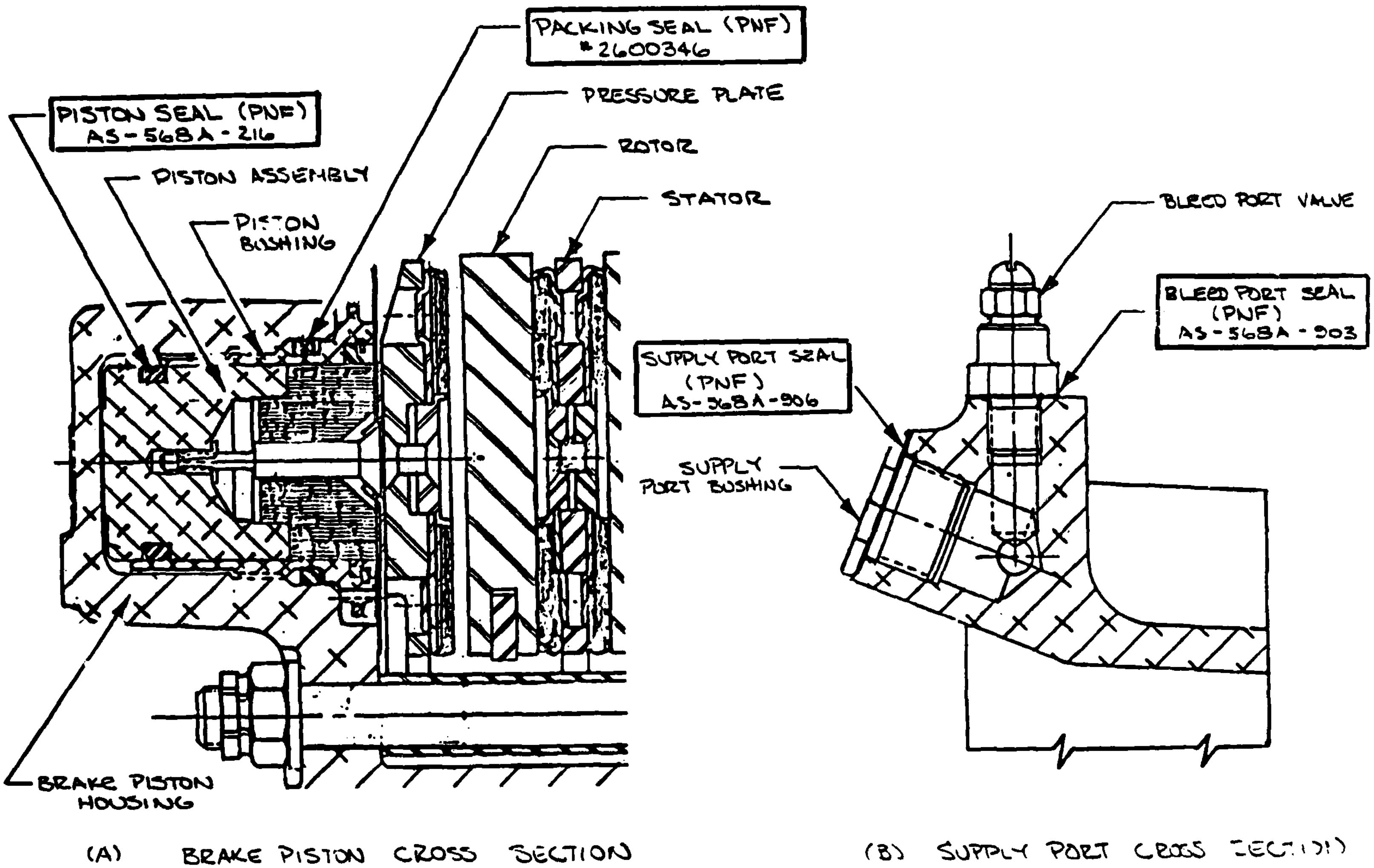


Figure 2.7 Brake Seals Exposed to AO-8 Fluid

MIL-H-5606 portion is performed by applying maximum brake pressure and cracking the reservoir bleed valve to circulate MIL-H-5606 through the brake system. The AO-8 portion is then serviced by opening the brake bleed valve and pumping AO-8 through the reservoir and brakes. After bleeding, additional AO-8 is added to fill the replenishment reservoir.

When the AO-8 portion of the system is serviced the deboost valve piston is bottomed against the end cap, the replenishment valve opened and the standpipe is in the plug cavity. AO-8 fluid is then pumped through the AO-8 fill valve into the replenishment reservoir, through the replenishment valve into the low pressure volume in the deboost valve, into the plug cavity, down the standpipe and into the brakes and out the brake bleed port. As fluid passes through the system any air in the system will be forced out through the brake bleed port. For example, air in the deboost valve rises and collects in the plug cavity. This air is forced down the standpipe and out the brake as the deboost valve volume fills with fluid.

Detailed fill and bleed procedures for both the MIL-H-5606 and the AO-8 portions of the two-fluid brake hydraulic system are given in Table 2.2.

2.4.5 SYSTEM SAFETY FEATURES

Several features have been included in the two-fluid brake hydraulic system configuration to improve system safety.

Four replenishment systems, one for each tandem-wheel-pair (Figure 2.4) have been included in the system design to prevent the loss of braking capability in the event of a failure. For example, if a hydraulic line between the deboost valve and brake were to burst, only the braking capability and fluid associated with that wheel pair and its replenishment system is lost. Normal braking capability and replenishment capacity is maintained on the other three paired wheel sets.

The replenishment reservoir is pressurized normally by pilot metered pressure and in the event of a failure by the copilot metered pressure through a shuttle valve. This configuration prevents the loss of replenishment capability when pilot metered pressure is lost.

TABLE 2.2

AO-8 TWO-FLUID BRAKE SYSTEM FILL AND BLEED PROCEDURE

The Fill and Bleed Procedures are Identical

1. Apply full brakes
   3000 psi is applied to deboost valve piston and replenishment revervoir piston.
2. Bleed replenishment reservoir MIL-H-5606 system.
3. Open brake assembly bleed valve.
   Deboost valve piston moves to bottom position and opens replenishment valve.
   Replenishment reservoir piston bottoms, left side.
4. Pump AO-8 into charge valve.
   AO-8 flows thru reservoir, deboost valve, and brake assembly purging air from system.
5. Close brake assembly bleed valve.
6. Release brake pedals.
   Return pressure is now applied to deboost valve piston and replenishment revervoir piston.
7. Pump AO-8 into charge valve until fluid level rod indicates "full".
   Pumping AO-8 first raises deboost valve piston 1/8" and closes replenishing valve.
   Pumping AO-8 then moves replenishment reservoir piston right to full position.
8. Close charge valve.
9. Apply full brakes and release after rod stops.
   Piston opens replenishment valve drawing AO-8 from reservoir to fill brake cylinders.
10. Repeat steps 7 and 8.

The use of pilot metered pressure to power the replenishment reservoir provides an additional safety feature unique to this configuration (see Section 2.4.2). If a high pressure air charged accumulator were used to power the reservoir, the brakes could be pressurized and locked due to leakage through the replenishment valve.

2.4.6 RELIABILITY AND MAINTAINABILITY

The normal reliability and maintainability of the two-fluid brake system are less due to the additional potential failure and leakage points in the system and the added servicing associated with the AO-8 fluid replenishment system. However, the fleet service cost of the system may be reduced due to the reduction in fire related costs. A detailed reliability and maintainability study will be performed during Task 6 of this contract.

2.4.7 WEIGHT PENALTY

The conversion of the KC-135 brake hydraulic system to the two-fluid configuration will increase the weight of the aircraft approximately 39 pounds. A weight breakdown is shown in Table 2.3. This estimate assumes that the size of the hydraulic lines from the deboost valve to the brakes are unchanged.

TABLE 2.3 TWO-FLUID BRAKE HYDRAULIC SYSTEM WEIGHT BREAKDOWN

QTY.	ITEM/DESCRIPTION	ESTIMATED WEIGHT PER ITEM (lbs)	TOTAL WEIGHT PER AIRCRAFT (lbs)
WEIGHT REDUCTIONS			
8	QUANTITY MEASURING FUSE	0.54	- 4.32
4	FLUID, MIL-H-5606, 70.43 IN3/2 WHEEL SET	2.26	- 9.03
WEIGHT INCREASES			
4	RESTRICTOR	.19	.76
2	RESERVOIR SHUTTLE VALVE	.83	1.66
4	REPLENISHMENT RESERVOIR WITH FITTINGS	5.23	20.92
4	DEBOOST VALVE MODIFICATIONS	--	--
4	*TUBING, RESERVOIR TO TO DEBOOST VALVE, 15 INCHES	.06	.24
2	*TUBING, PILOT METERED PRESSURE TO RESERVOIR SHUTTLE VALVE, 30 INCHES	.12	.24
2	*TUBING, CO-PILOT METERED PRESSURE TO RESERVOIR SHUTTLE VALVE, 30 INCHES	.12	.24
6	TEE FITTINGS	.04	.24
2	FLUID, MIL-H-5606, 5.00 IN3/GEAR	.16	.32
4	FLUID, AO-8, 101.26 IN3/2 WHEEL SET	6.83	27.34
6	*TUBING, SHUTTLE VALVE TO RESERVOIR, 5 INCHES	.03	.18
	TOTAL WEIGHT ADDED PER AIRCRAFT (APPROXIMATE)		38.79

* All tubing is 1/4 inch O.D., wall steel tube.

SECTION III

DYNAMIC ANALYSIS OF THE TWO-FLUID BRAKE HYDRAULIC SYSTEM FOR THE ASSESSMENT OF SYSTEM MODIFICATIONS

3.1 OBJECTIVE OF THE DYNAMIC ANALYSIS

The dynamic analysis of the KC-135 brake hydraulic system and the two-fluid brake hydraulic system was performed to determine (1) the effect which the proposed two-fluid system has upon the dynamic response of the brake hydraulic system, and (2) the modifications to the two-fluid system configuration which are required to achieve a dynamic response characteristic comparable to that of the KC-135 brake hydraulic system.

3.2 METHOD OF ANALYSIS

The KC-135 and the two-fluid brake hydraulic system were analyzed using the government owned Hydraulic System Frequency Response (HSFR) computer program. The systems were modelled and HSFR was used to determine the frequency response i.e., the gain and phase angle relationships between the pressure into the antiskid valve (input) and brake pressure (output) of each brake system. These frequency responses were then compared to analyze the effects of the two-fluid configuration on the brake system dynamic response.

Several minor program modifications were made to (1) model the two-fluid isolation unit (deboost valve), (2) include two types of hydraulic fluid within a single system, and (3) model the brake. In addition the fluid characteristics (density, bulk modulus, and viscosity) of the AO-2 and AO-8 fluids* were input in the program. These program changes are discussed in Appendix A-1.

The general procedure employed to analyze the two-fluid system configuration is outlined below.

*The AO-8 fluid is identical to the AO-2 fluid with the addition of a viscosity index improver.

1. A HSFR computer model of the KC-135 brake hydraulic system from the antiskid valve to the brake was formed.

2. The frequency response generated with the HSFR KC-135 brake hydraulic system computer model was compared with actual laboratory data. Key parameters within the computer model were varied to improve the correlation between the computer model and laboratory results. When satisfactory correlation was obtained the configuration was fixed and a final frequency response analysis performed. The final computer model configuration and the associated frequency response will be termed the baseline system and the baseline frequency response respectively throughout this section.

3. The baseline system computer model was converted to the two-fluid configuration and a frequency response analysis performed. The two-fluid configuration employs MIL-H-5606 fluid from the antiskid valve to the deboost valve and AO-8 from the deboost valve to the brake while the baseline system uses MIL-H-5606 throughout the system.

4. The fluid properties (bulk modulus, density, and viscosity) were varied one at a time to determine how each property affects the dynamic response of the brake hydraulic system.

5. The configuration of the two-fluid brake hydraulic system was changed to determine the system modifications which are required to achieve dynamic response comparable to the baseline system.

3.3 HSFR COMPUTER MODEL OF THE KC-135 BRAKE HYDRAULIC SYSTEM

The KC-135 brake hydraulic system was modelled from the antiskid valve to the brake. The HSFR KC-135 brake system computer model along with a diagram of the actual KC-135 system are shown in Figure 3.1. The KC-135 brake system configuration is defined in Table 3.1 while the HSFR computer model is defined

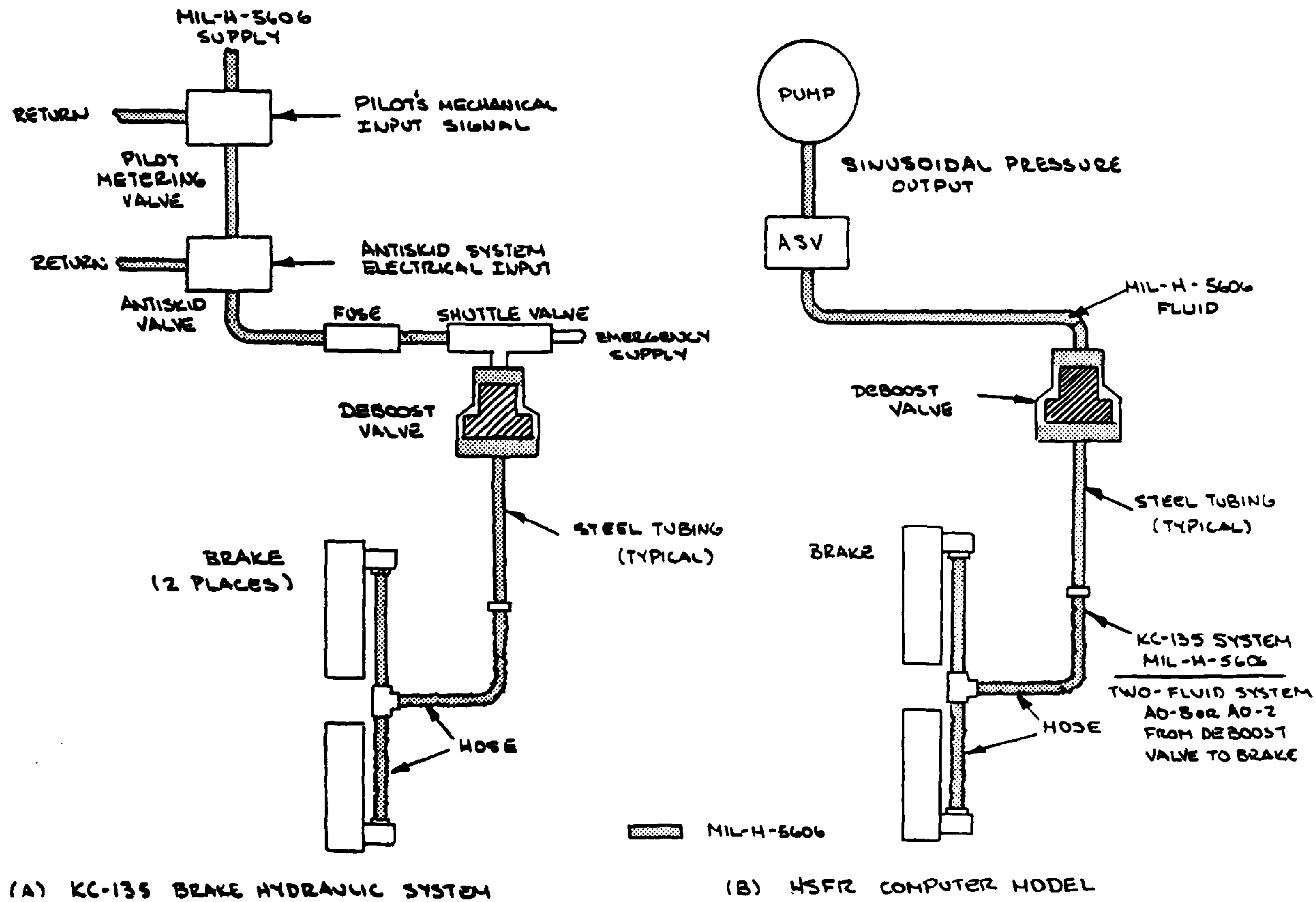
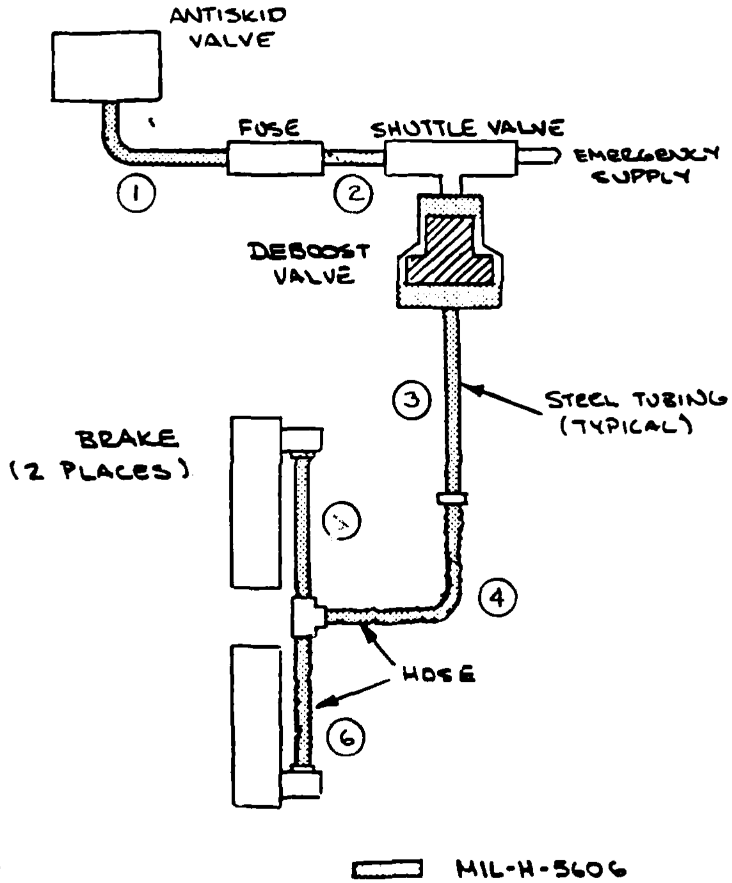


Figure 3.1 KC-135 Brake Hydraulic System Computer Model

TABLE 3.1 KC-135 BRAKE HYDRAULIC SYSTEM CONFIGURATION

SYSTEM CONFIGURATION FROM THE ANTISKID VALVE TO THE BRAKE

DESCRIPTION	LINE NUMBER	LINE * SIZE	LINE LENGTH (INCHES)
ANTISKID VALVE TO FUSE	1	8S49	11
FUSE TO SHUTTLE	2	8S49	16
DEBOOST TO HOSE	3	8S49	170
HOSE	4	1/2 IN.	81
BRAKE LINE	5	6S35	63
BRAKE HOSE	6	3/8 IN.	24



* LINE SIZE
TUBING DESIGNATION

8 S 49
- WALL THICKNESS (i.e., 0.049 IN)
- STAINLESS STEEL
- O.D. IN 1/16 ths (i.e., 8/16 IN)

in Table 3.2. The computer model, as shown in Figure 3.1, contains a pump, antiskid valve, hydraulic lines and hoses, deboost valve, and brakes.

The pump models the brake hydraulic supply system from the pilot metering valve to the antiskid valve. It provides a pressure and flow source for the brake system and defines the test or run conditions to be analyzed.

The fuse and shuttle valve found in the actual KC-135 system were not modelled as individual components. Their effects are included in the antiskid valve component. This approach was taken for two reasons; (1) there was insufficient engineering data to define the characteristics of the individual components and (2) it was determined through a series of computer runs that the antiskid valve, shuttle valve and fuse could be lumped together without significantly changing the response of the overall system.

The deboost valve was modelled with two short-large diameter lines, a volume, and a differential area piston with mass and damping. This approach to modelling the deboost valve includes the effects of the fluid volume found at each end of the deboost valve, the elasticity of the container, the effect of the differential piston area on pressure and flow, the inertial effect of the piston mass and the friction associated with the piston seals. A new HSFR hydraulic component subroutine was developed to model the pressure/flow characteristics associated with the deboost valve piston. Details of the new component are given in Appendix A-1.

The brake was modelled with a modified version of the HSFR accumulator subroutine. The accumulator subroutine was modified so that the mechanical spring stiffness of the brake could be input directly rather than calculated as a accumulator air spring stiffness term. With this version of the accumulator subroutine the inertia (mass), seal friction, stiffness, brake housing elasticity, and fluid volume characteristics of the brake were modelled. The change made to the accumulator subroutine is outlined in Appendix A-1.

The lines and hoses associated with the brake system were modelled with the standard HSFR line and hose subroutine.

TABLE 3.2 HSFR BRAKE HYDRAULIC SYSTEM COMPUTER MODEL

DESCRIPTION	ITEM NUMBER	HSFR MODEL	HSFR PARAMETERS
PUMP	1	EMPIRICAL PUMP	PUMP PRESSURE = 200 PSI
ANTISKID VALVE	2	VALVE	VALVE GAIN = 25 PSI/CIS
A/S VALVE TO DEBOOST VALVE	3	LINE*	LINE LENGTH = 27.0 IN O.D. = .5 IN WALL THICKNESS = .049 IN
DEBOOST VALVE	4	LINE*	LINE LENGTH = 1.62 IN O.D. = 2.57 IN WALL THICKNESS = .22 IN
		PISTON	AREA 1 = 3.533 IN**2 AREA 2 = 11.027 IN**2 MASS = .002 LBS*SEC**2/IN DAMPING = .2 LBS*SEC/IN
		VOLUME	VOLUME = 10.58 IN**3
		LINE*	LINE LENGTH = .4 IN O.D. = 4.15 IN WALL THICKNESS = .19 IN
DEBOOST VALVE TO HOSE	5	LINE*	LINE LENGTH = 170.0 IN O.D. = .5 IN WALL THICKNESS = .049 IN
HOSE	6	HOSE**	LINE LENGTH = 81.0 IN I.D. = .5 IN
BRAKE LINE	7	LINE*	LINE LENGTH = 63.0 IN O.D. = .375 IN WALL THICKNESS = .035 IN
BRAKE HOSE	8	HOSE**	LINE LENGTH = 24.0 IN I.D. = .375 IN
BRAKE	9	ACCUMULATOR*	PISTON MASS = 3.944 LBS*SEC**2/IN PISTON RADIUS = 1.95 IN WALL THICKNESS = .25 IN LENGTH = .46 IN STIFFNESS = 276750 LBS/IN DAMPING = 1483 LBS*SEC/IN

* MODULUS OF ELASTICITY = 28000000 PSI
** BULK MODULUS = 16000 PSI

The fluid properties (bulk modulus, density, and viscosity) of MIL-H-5606, AO-8, and AO-2 installed in the HSFR program are listed in Appendix A.

3.3.1 CORRELATION OF THE COMPUTER MODEL WITH TEST DATA

A series of preliminary runs were made to adjust key model parameters and correlate the frequency response generated with the computer model with actual laboratory test data. Prior to the correlation effort the parameters within the model which could logically be varied were identified. These parameters are:

- Antiskid Valve Gain (VG)
- Deboost Valve Piston Seal Friction (BR)
- Hose Elasticity (BH)
- Brake Mass (M)
- Brake Piston Seal Friction (B)
- Brake Stiffness (K)

The values of these parameters could not be accurately defined prior to the correlation effort. System parameters (such as line length, diameter and wall thickness, material properties, volume, piston mass, etc.) having values which are defined by the configuration geometry or could be accurately calculated were not varied.

During the correlation effort it was found that the frequency response of the brake hydraulic system is controlled by antiskid valve gain (VG), hose elasticity (BH), brake natural frequency ($\omega = \sqrt{K/M}$) and the brake damping ratio ($\zeta = B/(2\sqrt{KM})$). Deboost valve piston seal friction (BP) was found to have little effect on the system frequency response. The values of the key parameters determined during the correlation effort along with system configuration input data are listed in Table 3.2.

The computer generated frequency response between 1.0 and 40.0 Hertz of the KC-135 brake hydraulic system baseline model at 643 psi brake pressure is shown in Figure 3.2. Excellent correlation between the model and laboratory data at brake pressures of 321 psi and 643 psi is exhibited in Figures 3.3 and 3.4. The response of the actual KC-135 system at the 321 ± 200 psi and 643 ± 200 psi along with the model data are plotted in these figures. Three response curves corresponding to three different antiskid valves of identical make and model, are shown in the figures to illustrate the variation in system response which is obtained simply by replacing the valve in the system with an identical unit.

3.4 RESULTS OF THE TWO-FLUID SYSTEM ANALYSIS

The baseline KC-135 brake hydraulic system computer model was converted to the two-fluid system configuration and frequency response analyses performed to determine the effect which the two-fluid system has upon the dynamic response of the brake hydraulic system. The two-fluid system configuration is shown in Figure 3.5. The two-fluid system utilizes MIL-H-5606 hydraulic fluid between the antiskid and deboost valves and AO-8 fluid between the deboost valve and brake, while the baseline system uses MIL-H-5606 fluid throughout the entire system.

3.4.1 EFFECT OF THE TWO-FLUID SYSTEM CONFIGURATION ON PERFORMANCE

The system frequency response of the two-fluid configuration at 643 psi brake pressure along with the baseline response is shown in Figure 3.6. The phase angle (time lag) of the brake hydraulic system is increased by conversion to the two-fluid configuration.

3.4.2 EFFECT OF FLUID PROPERTIES ON SYSTEM RESPONSE

A series of computer runs were conducted to determine what fluid property(s) caused the change in system frequency response that was observed when the baseline system (MIL-H-5606 fluid only) was converted to the two-fluid

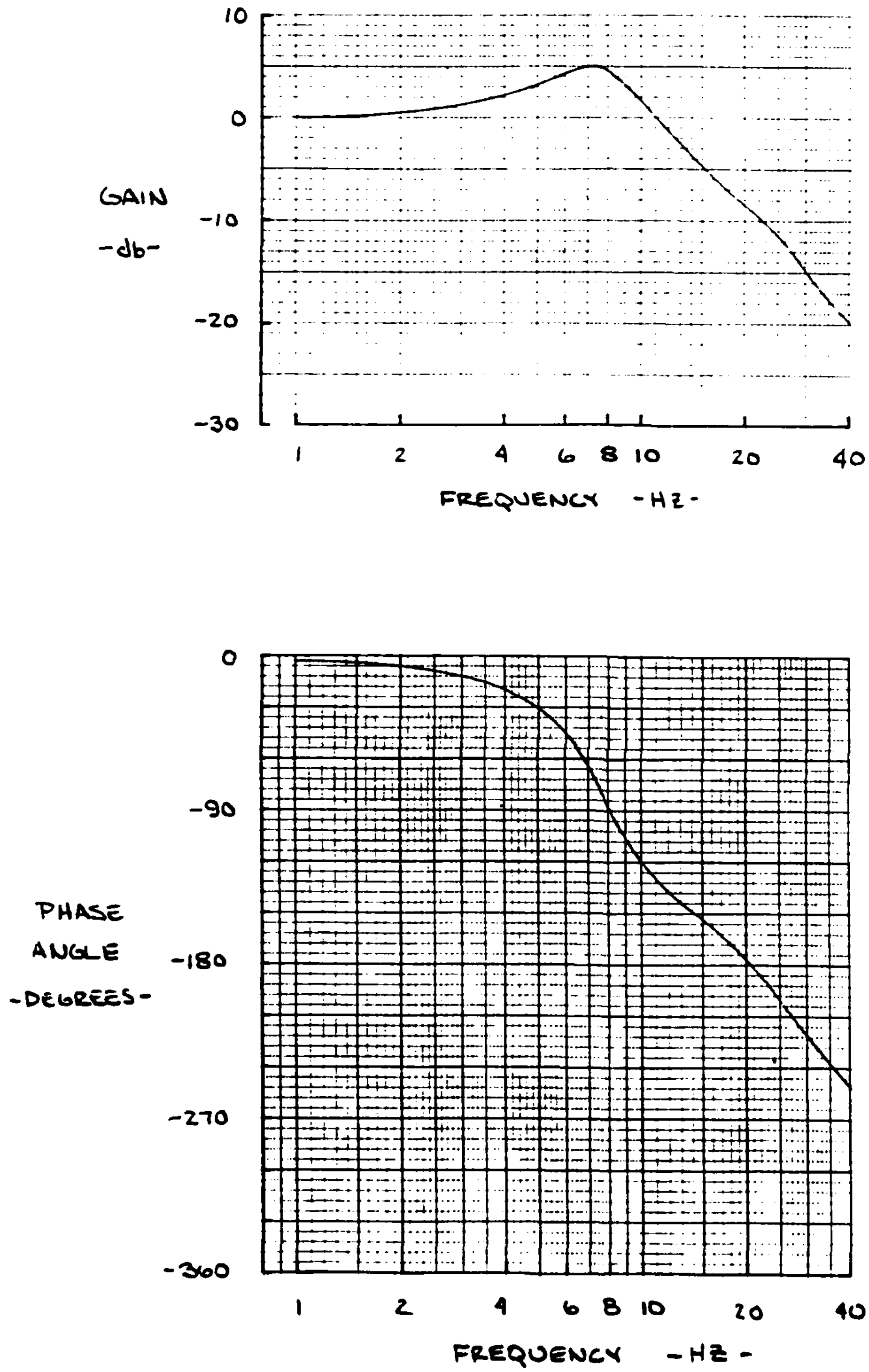
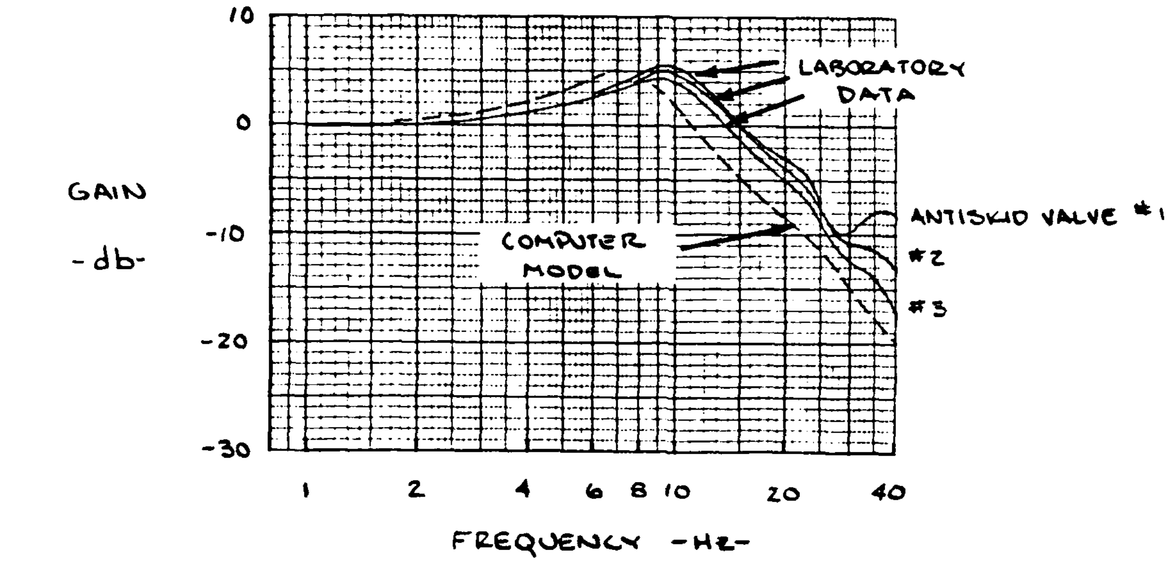


Figure 3.2 KC-135 Brake Hydraulic System Frequency Response, Computer Model



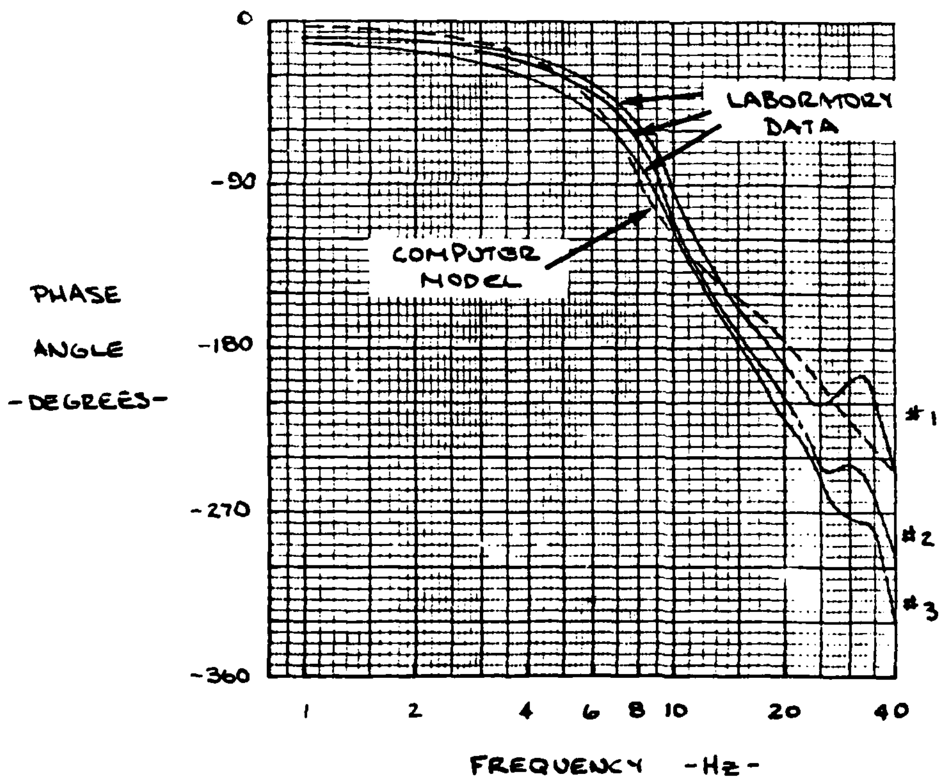


Figure 3.3 Model Correlation at 643 PSI (Brake Pressure)

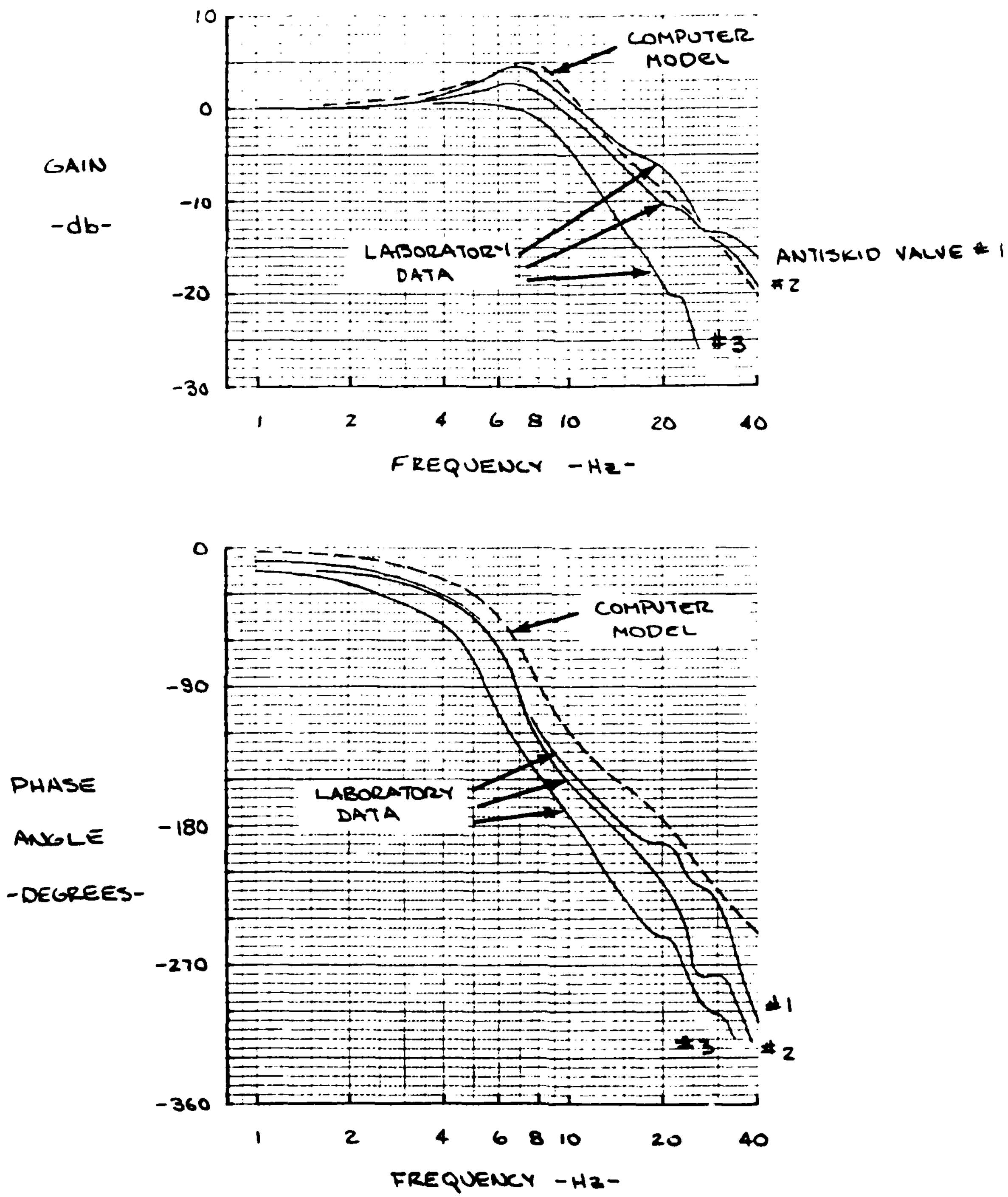


Figure 3.4 Model Correlation at 321 PSI (Brake Pressure)

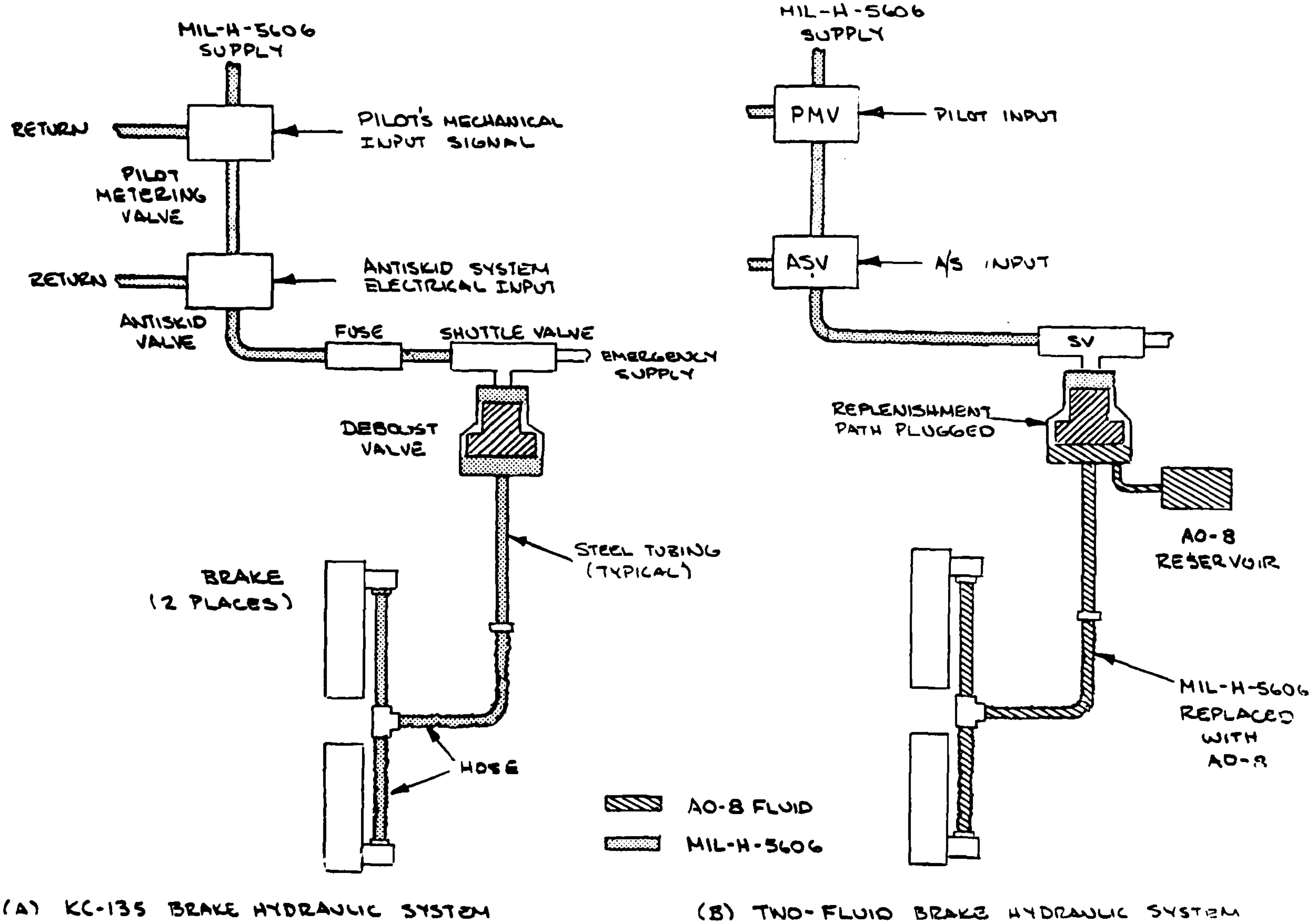
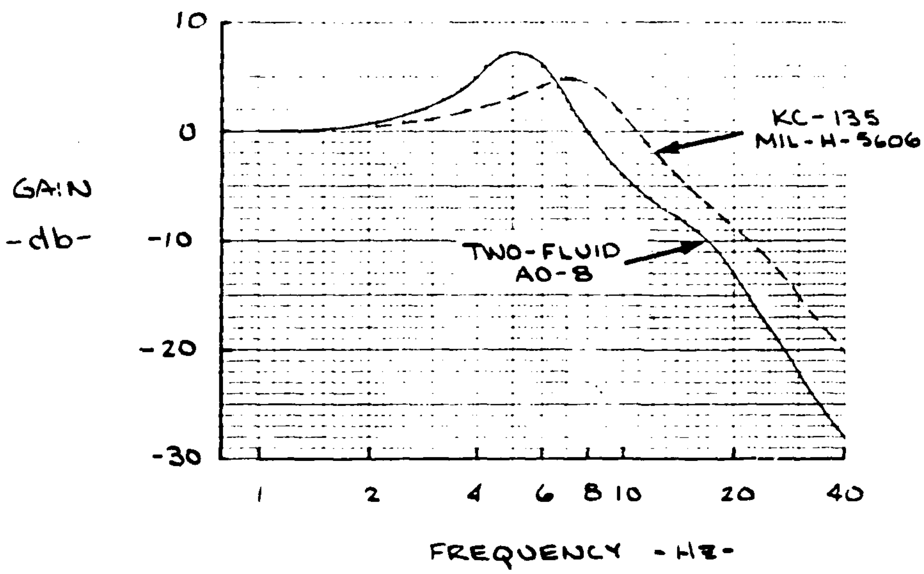


Figure 3.5 Two-Fluid Brake Hydraulic System



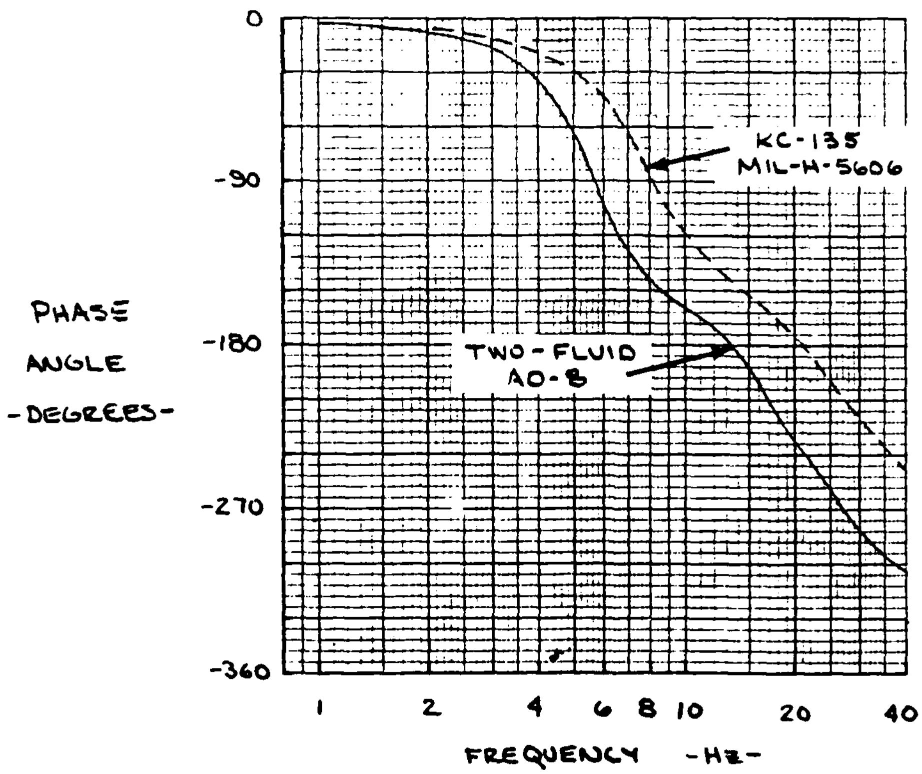


Figure 3.6 Effect of the Two-Fluid Configuration on System Frequency Response

configuration (combined MIL-H-5606 and AO-8 fluid system). The fluid properties (bulk modulus, viscosity, and density) were varied one at a time to determine the effect of each on the system response. The results of this effort are shown in Figures 3.7, 3.8, and 3.9. The fluid properties associated with each test condition are given in Table 3.3. Density was found to be the key parameter responsible for the increase in the phase angle (Figure 3.9). Viscosity and bulk modulus have only a minor effect.

3.4.3 AO-2 VERSUS AO-8

The frequency response of the two-fluid system configuration with AO-2 and AO-8 was compared. The AO-2 is a less viscous version of the AO-8 fluid. The frequency response of the system with AO-2 and AO-8 is shown in Figure 3.10. The AO-2 and AO-8 fluid properties use in this analysis are given in appendix A. The system phase angle is reduced slightly at low frequency when the less viscous AO-2 is used instead of the AO-8 fluid. However, this reduction in phase angle does not represent a significant improvement in system performance.

3.4.4 SYSTEM CONFIGURATION MODIFICATIONS

A series of computer runs were performed to determine the two-fluid brake hydraulic system configuration modifications which would be required to match the frequency response of the baseline system. The system configuration parameters which can logically be varied are (1) the antiskid valve gain, (2) hose length, and (3) tubing/hose diameter. The effect of these parameters on the system frequency response is shown in Figures 3.11, 3.12, and 3.13. Changing the antiskid valve gain does not shift the system phase angle. However, replacing all hoses in the system with solid lines or increasing the line diameters does shift the phase angle of the two-fluid system toward the baseline.

While increasing the diameter of the hydraulic lines does shift the system phase angle toward the baseline, it also represents a weight penalty. To match the performance of the baseline system the cross sectional area of the hydraulic lines and hoses must be doubled (the ratio of fluid density to area

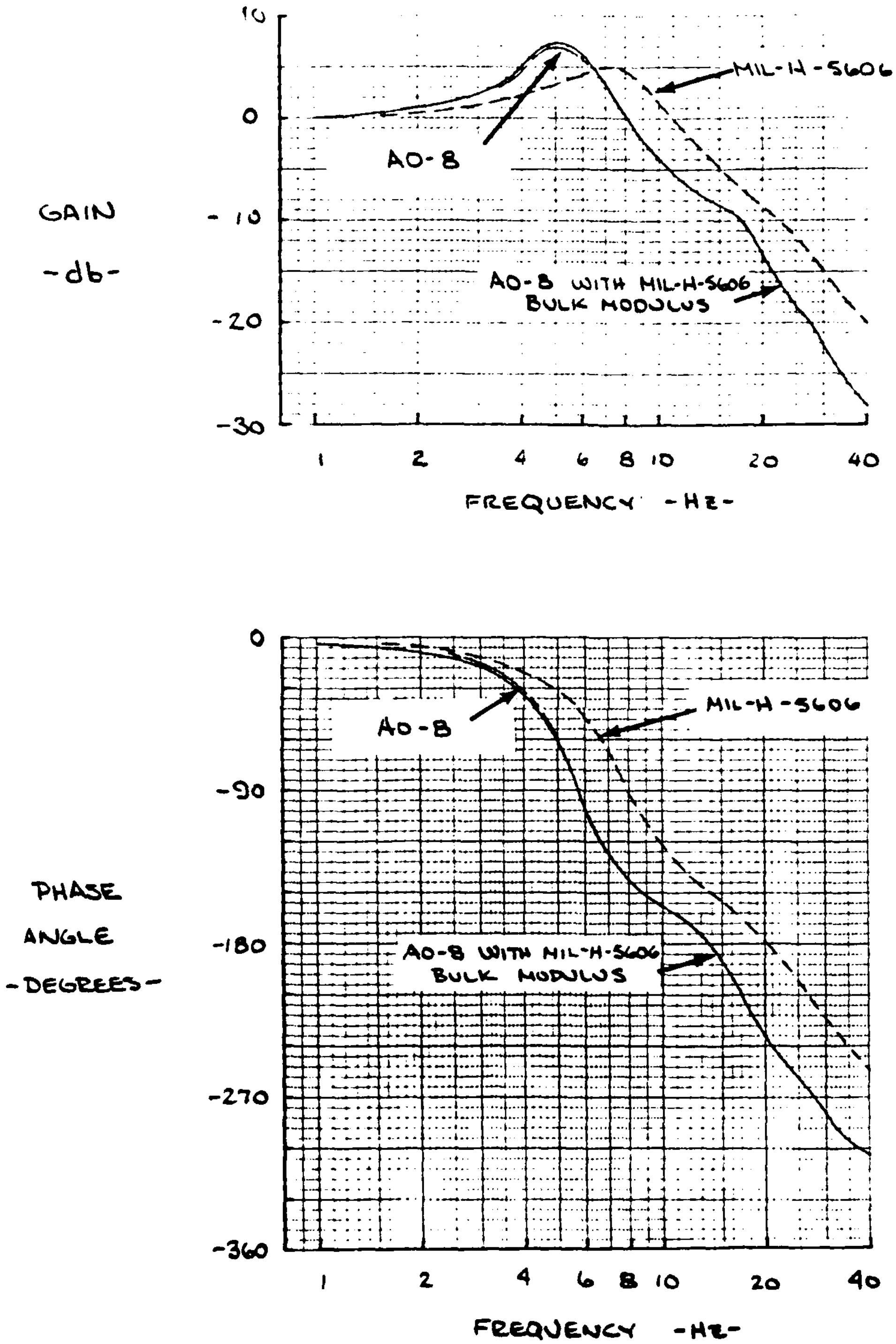


Figure 3.7 Effect of Fluid Bulk Modulus on Frequency Response

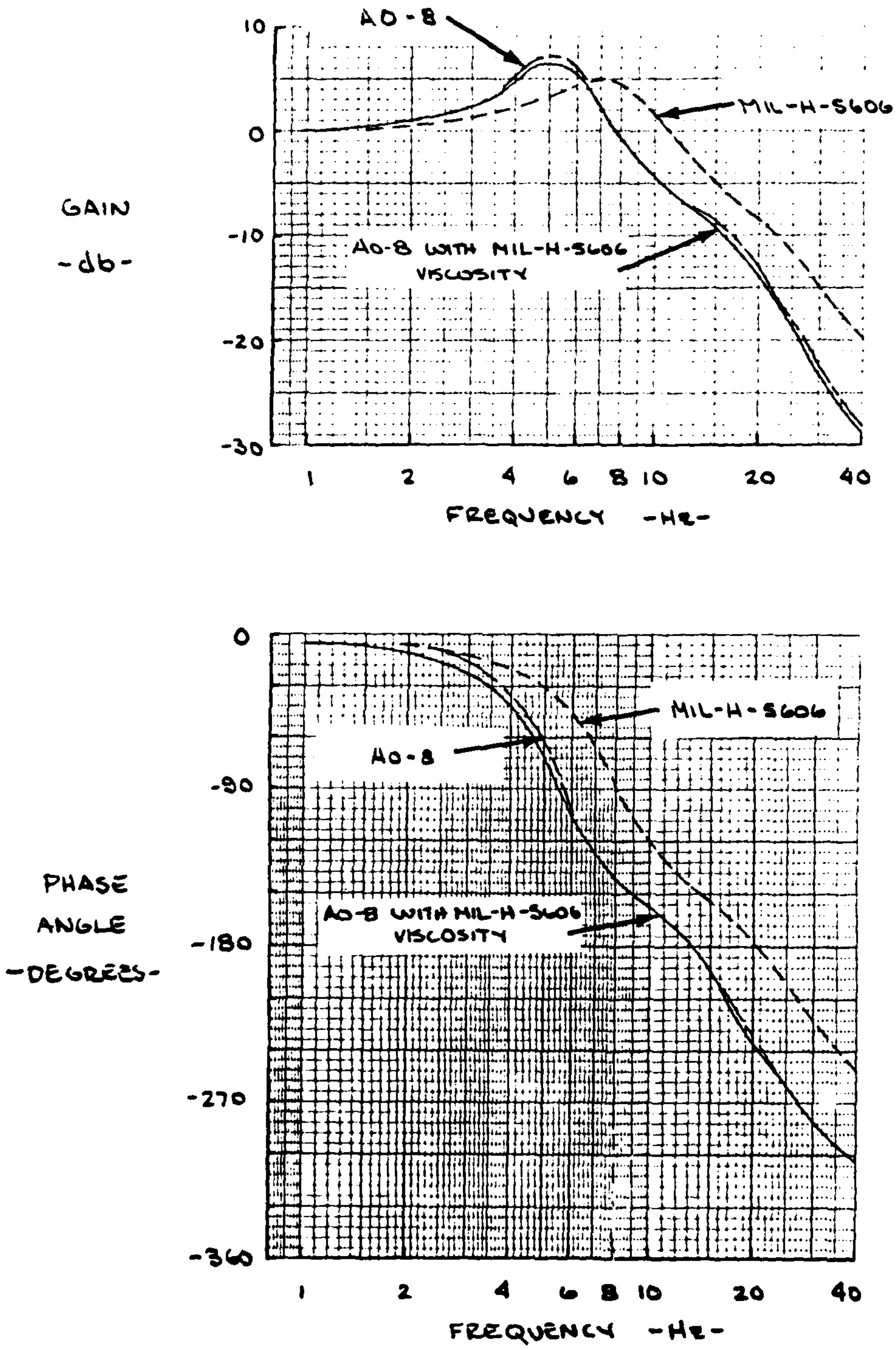


Figure 3.8 Effect of Fluid Viscosity on Frequency Response

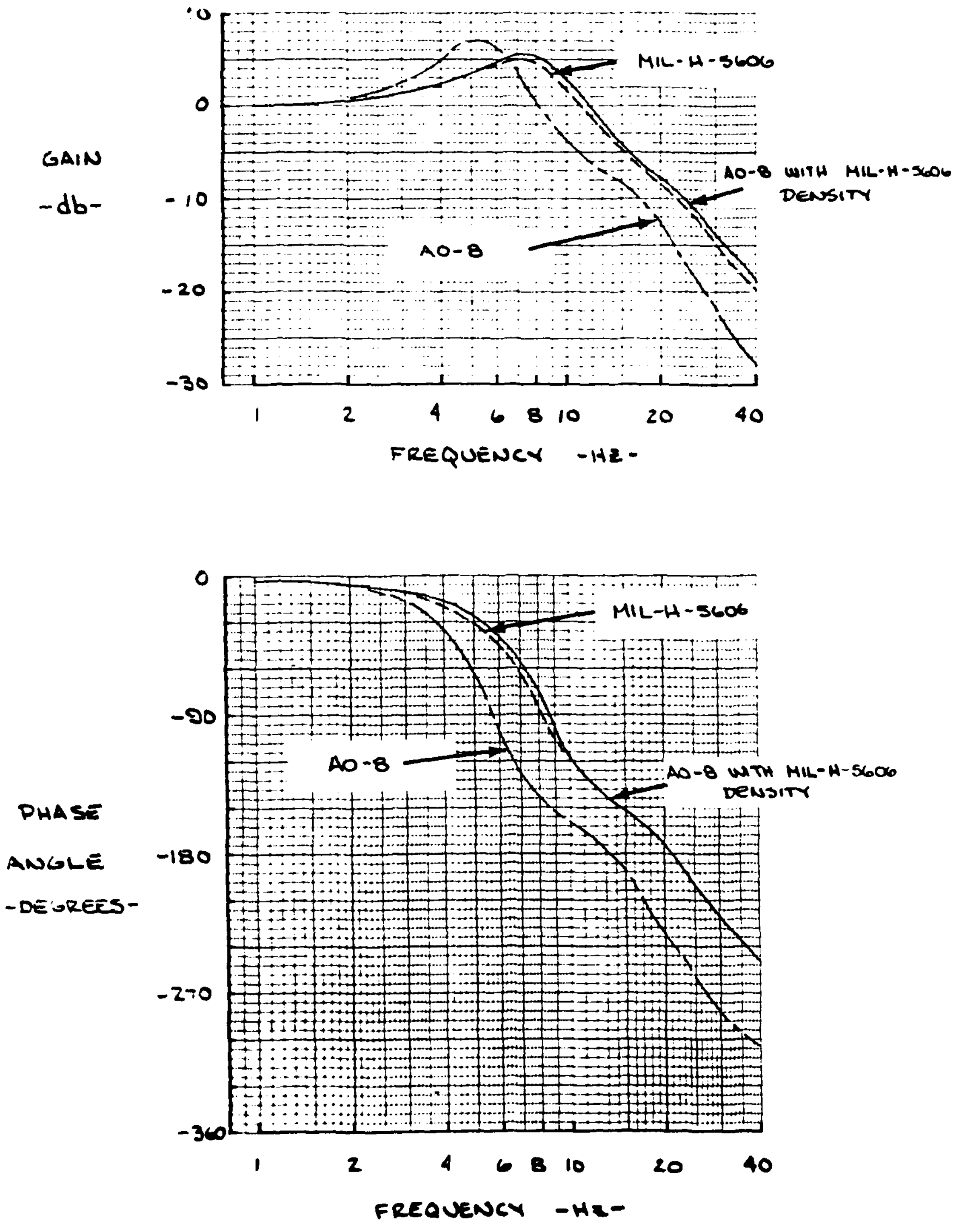
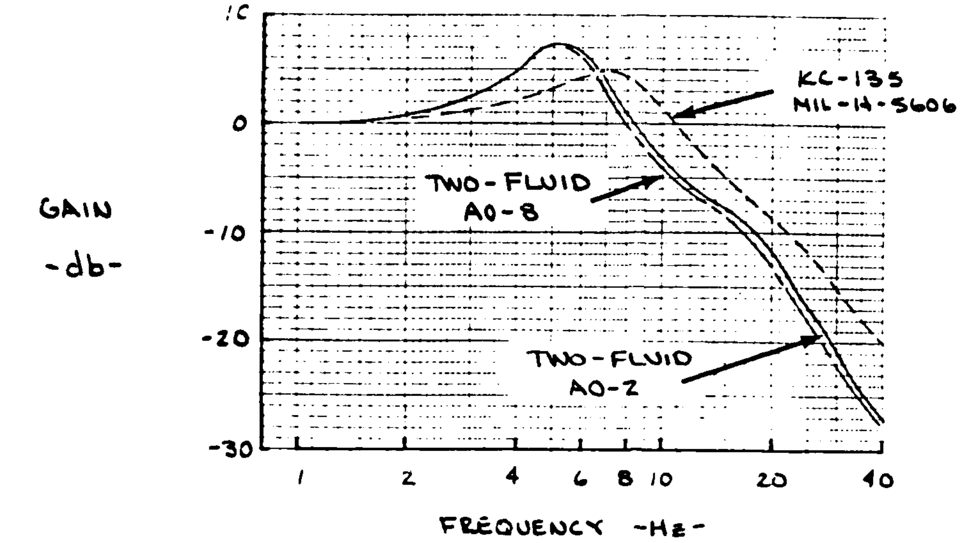


Figure 3.9 Effect of Fluid Density on Frequency Response

TABLE 3.3 FLUID PROPERTY SENSITIVITY TESTS

TEST	FLUID PROPERTIES USED IN TEST (SEE TABLE A.1 FOR NUMERICAL DATA)		
	BULK MODULUS	VISCOSITY	DENSITY
BULK MODULUS TEST	MIL-H-5606	AO-8	AO-8
VISCOSITY TEST	AO-8	MIL-H-5606	AO-8
DENSITY TEST	AO-8	AO-8	MIL-H-5606



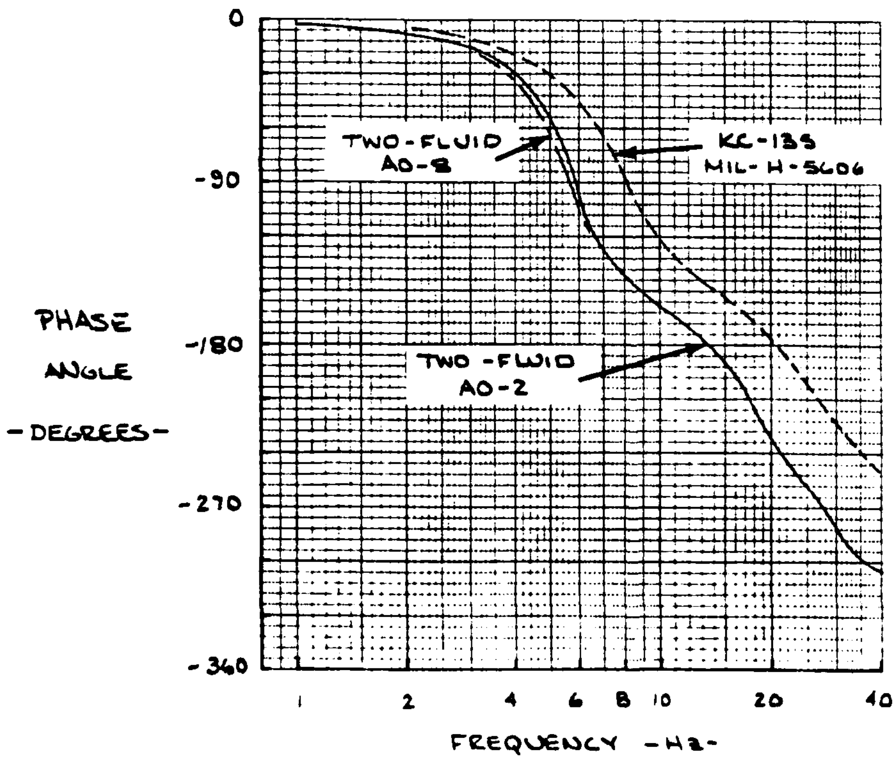


Figure 3.10 Frequency Response with AO-2 Versus AO-8

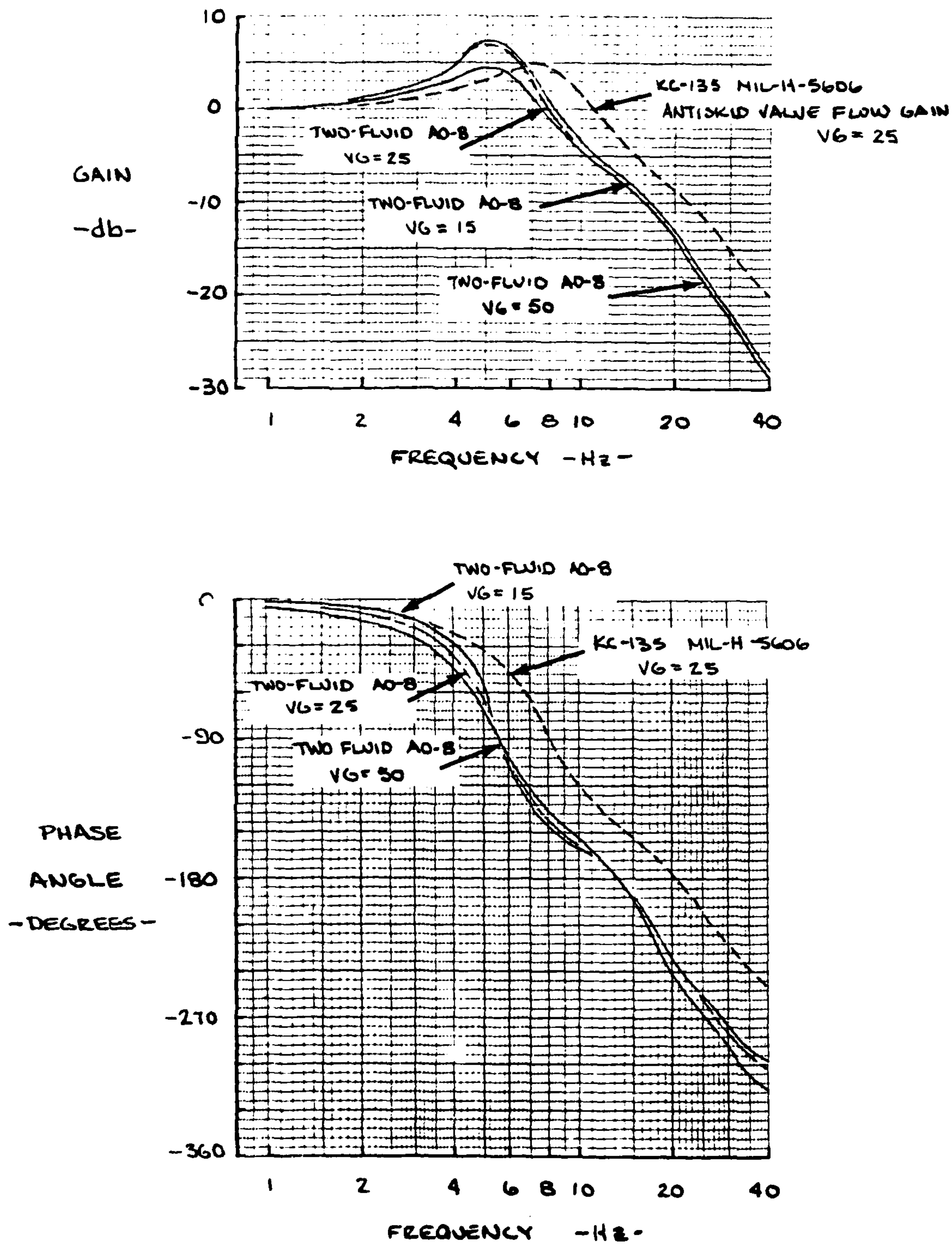


Figure 3.11 Effect of Antiskid Valve Flow Gain on Frequency Response

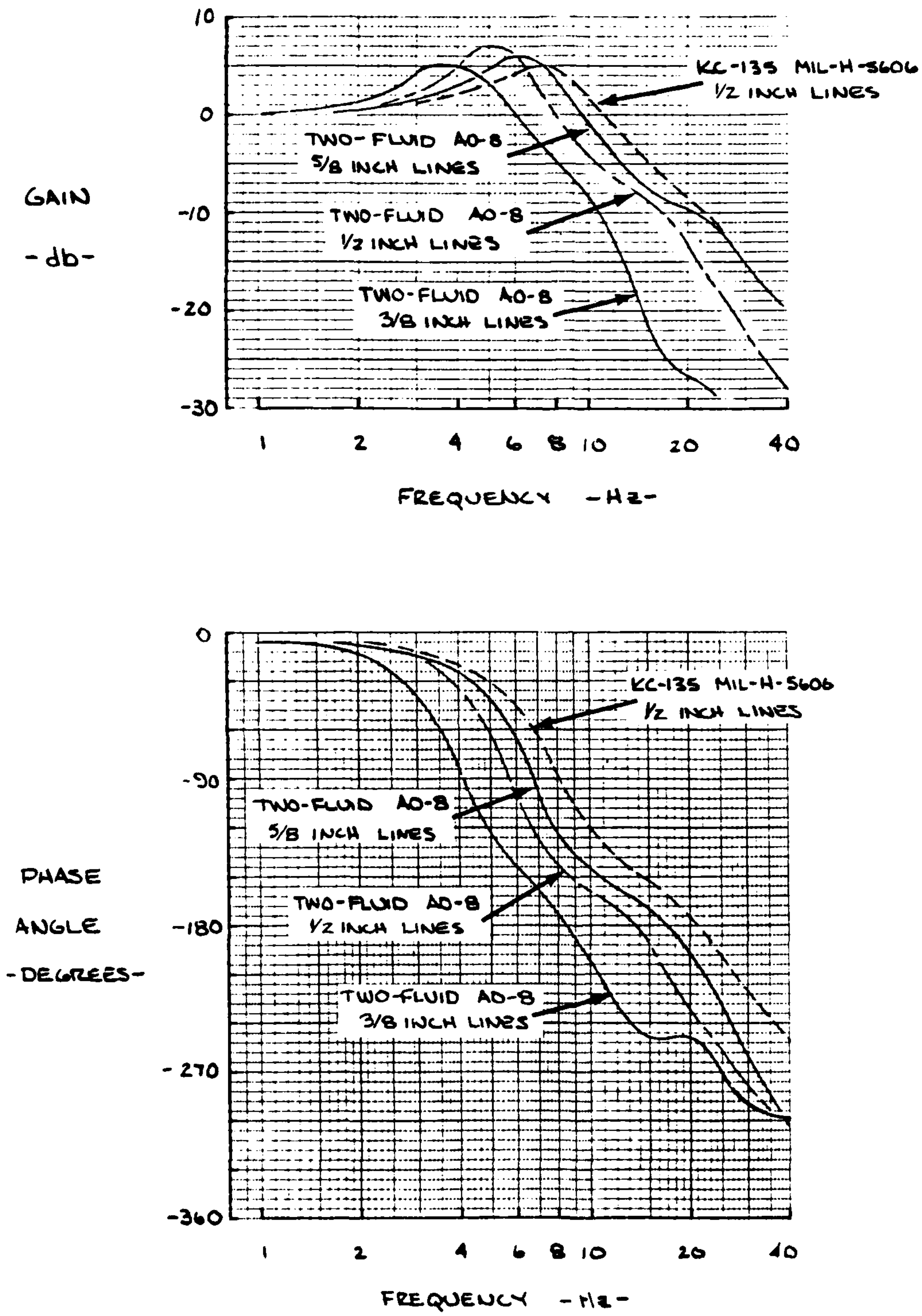
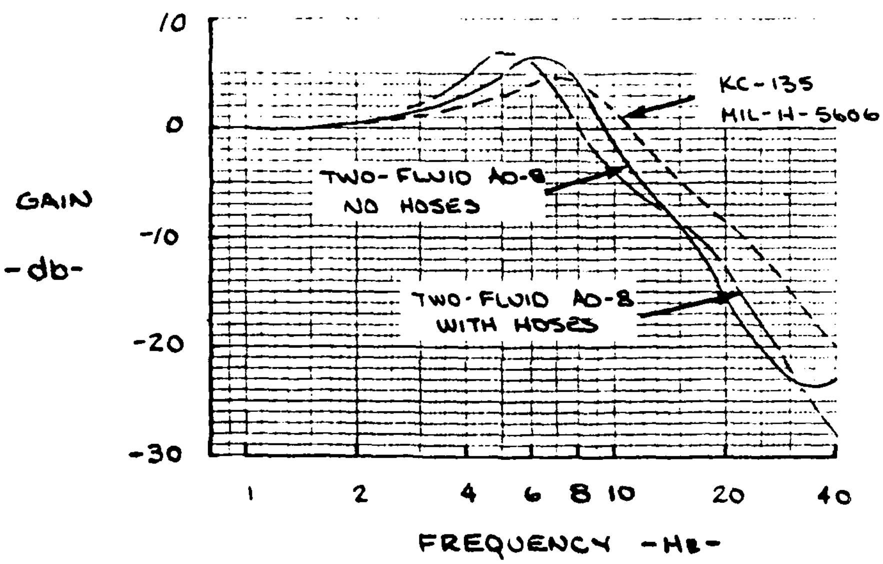


Figure 3.12 Effect of Line Diameter on Frequency Response



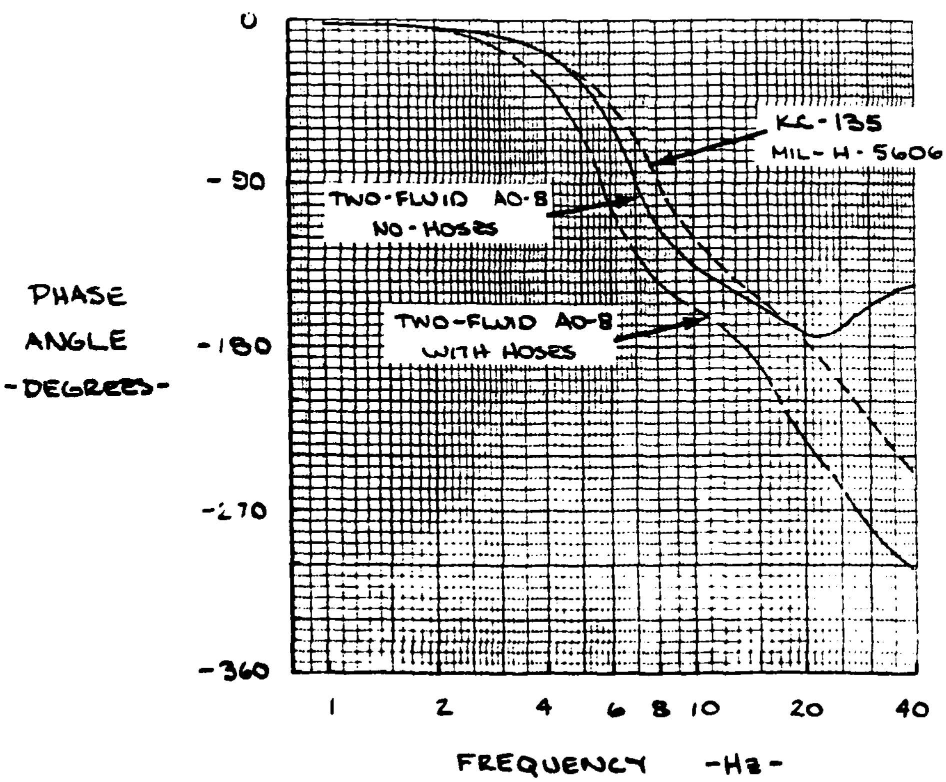


Figure 3.13 Effect of Hose Elasticity on Frequency Response

must remain constant). Doubling the area along with the 2.11 increase in density (AO-8 versus MIL-H-5606) results in a four fold increase in the weight of the fluid from the deboost valve to the brake as shown below. This weight penalty may be reduced and still shift the phase angle toward the baseline by replacing the hoses with solid lines.

Baseline System; MIL-H-5606	9.0 lbs/aircraft
Two-Fluid System; MIL-H-5606 (Standard diameter solid lines)	18.0 lbs/aircraft
Two-Fluid System; MIL-H-5606 and A08 (Increased line diameter)	36.0 lbs/aircraft

3.5 RECOMMENDATIONS FOR STUDY FLUID AND SYSTEM CONFIGURATION

3.5.1 STUDY FLUID

Results of the hydraulic system analysis indicate that density is the only fluid property which, when varied, significantly affects the dynamic response of the brake hydraulic system. Changes in the fuid kinematic viscosity have virtually no effect on the system dynamic response. Consequently, the dynamic response of the two-fluid system with the AO-8 fluid, and the two-fluid system with the AO-2 fluid, are nearly identical (the value of viscosity being the only difference between the two fluids).

Also, it is Boeing's understanding that viscosity is the only major fluid property that can be altered in the formulation of a chlorotrifluorethylene (CTFE) based fluid, bulk modulus and density being invariant.

Since analysis has shown the brake system dynamic response to be insensitive to variations in viscosity, this parameter and others may be selected at the discretion of the Air Force. Therefore, Boeing places no fluid property requirements on the formulation of the CTFE based fluid to be supplied to Boeing as GFE by AFWAL as specified in the subject contract.

3.5.2 STUDY CONFIGURATION

Although the two-fluid (AO-8) brake hydraulic system exhibits more phase lag (increased phase angle), it is Boeings opinion that this difference will not effect the braking performance of the aircraft. The primary operating mode of the antiskid system is of low frequency content (1/2 to 2 Hertz) occuring during the slow application of brake pressure between wheel skids. In the low frequency regime (Figure 3.6) the gain and phase angle of the brake hydraulic system with AO-8 and with MIL-H-5606 are virtually the same. Therefore, it appears that the performance will not be affected by use of the two-fluid/AO-8 brake hydraulic system configuration. Thus, it is anticipated that no system modifications to the two-fluid configuration will be necessary specifically to adjust braking performance.

SECTION IV

RECOMMENDED COMPONENT MODIFICATIONS FOR TESTING

The modifications to the KC-135 brake system which are required to form a complete and functional two-fluid brake hydraulic system have been described in Section II. However, for the purposes of this study it is necessary to make only those modifications which are active in the control portion of the brake hydraulic system. The AO-8 fluid replenishing components do not function during normal braking activity, and therefore will not be tested.

The recommended deboost valve and brake modifications which are required for testing are described below.

4.1 RECOMMENDED DEBOOST VALVE MODIFICATIONS

The modifications to the KC-135 deboost valve (Federal Stock Number 1650-00-570-8397) and added AO-8 fluid replenishment system which are necessary to convert the KC-135 brake system to a complete and functional two-fluid system are detailed in Section 2.4.2. However, to determine the effect of the two-fluid configuration on brake system performance it is not necessary to include the replenishment system. Consequently, it is recommended that all deboost valve modifications as specified in Section 2.4.2 be made with the exception of those modifications associated with the AO-8 fluid replenishment system. Thus, the following modifications are recommended:

(1) Plug the original deboost valve replenishment system flow path.

(2) Fabricate a new end cap and standpipe including the threaded hole for the AO-8 replenishment valve.

(3) Install PNF seals in areas with AO-8 fluid.

The fabrication and inclusion of the replenishment valve, replenishment reservoir, the pilot metered pressure line to the reservoir, and associated hardware are not needed. A simple needle valve will be installed in the replenishment valve threaded hole so fluid can be pumped through the deboost valve and brake system to accommodate filling and bleeding the AO-8 fluid volume. The needle valve will be closed during all tests. The recommended brake hydraulic system test configuration is shown in Figure 4.1.

4.1.1 REPLACEMENT DEBOOST VALVE PARTS

The deboost valve parts that will be used, modified or fabricated are listed in Table 4.1. The parts may be identified with the aid of Figure 2.5.

4.1.2 FILL AND BLEED PROCEDURE FOR TESTING

The AO-8 fluid fill and bleed procedure (Section 2.4.4) is modified to accommodate the elimination of the AO-8 replenishment system. The recommended fill and bleed procedure to be used during the component and system tests is detailed in Table 4.2.

4.1.3 PROCUREMENT AND MODIFICATION OF THE DEBOOST VALVE

The recommended modifications will be made to a new deboost valve assembly not exposed to MIL-H-5606 or other hydraulic fluid in compliance with the Air Force requirement stated in paragraph 4.8 of the SOW. The new deboost valve will be otained from the manufacturer; DECOTO Aircraft Inc., Yakima, Washington. This unit will be converted to a fluid isolator for use in the two-fluid brake hydraulic system. New components along with parts from the new deboost valve will be assembled for subsequent component and system testing. Prior to assembly each part will be cleaned using the procedure outlined in Section 4.3.

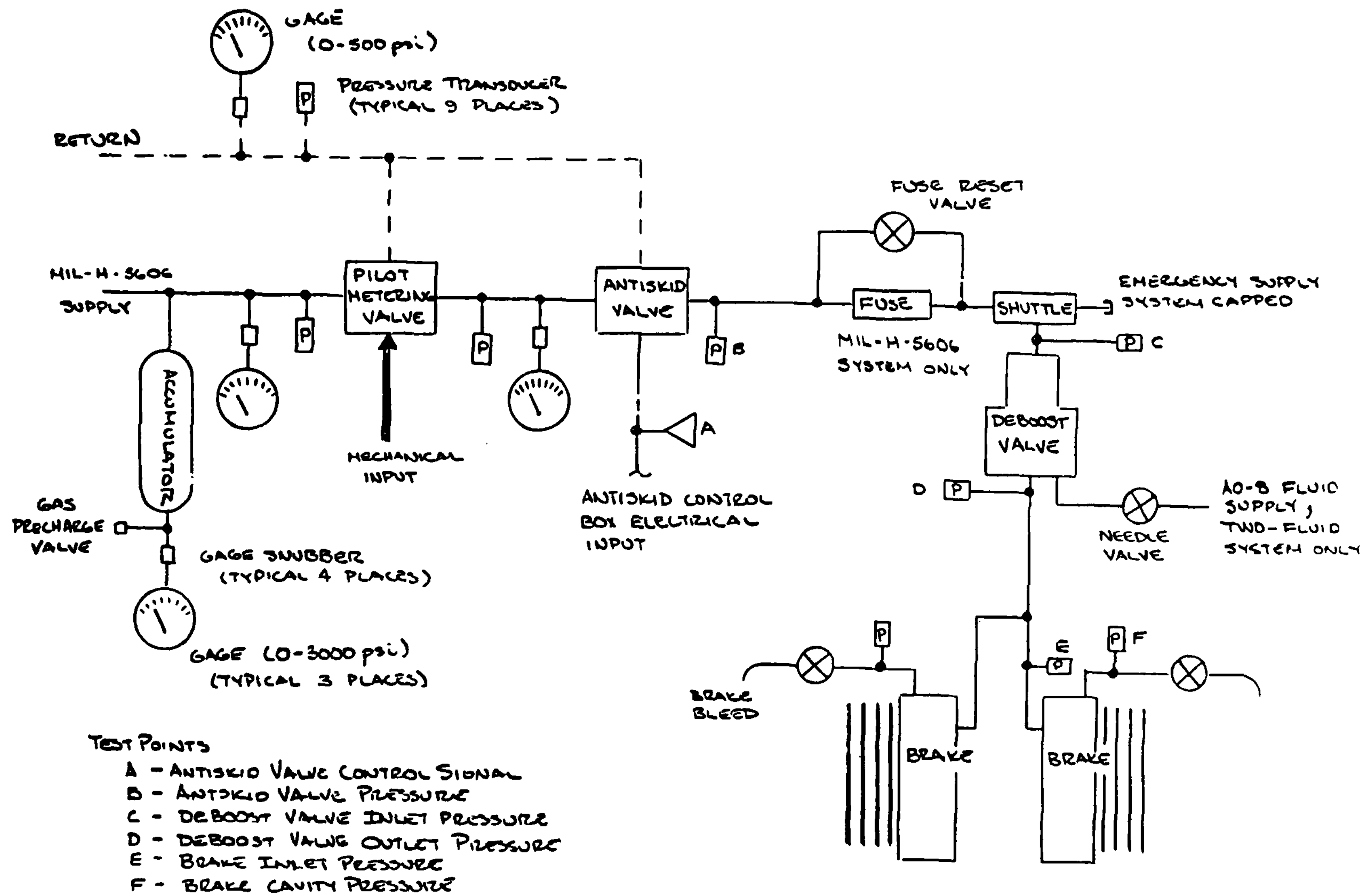


Figure 4.1 Two-Fluid Brake Hydraulic System Test Configuration

TABLE 4.1 DEBOOST VALVE MODIFICATIONS

ITEM	PART * NUMBER	MODIFICATION
PISTON	5-96372	MACHINE ORIGINAL REPLENISHMENT VALVE THREADS SMOOTH
BODY	6-96373	NO MODIFICATIONS
END CAP	5-96374	NOT USED
REPLENISHMENT PIN	6-83806-1	NOT USED
REPLENISHMENT VALVE ASSEMBLY	9-65813	NOT USED
LOCK RING	6-83876	NO MODIFICATIONS
NEW END CAP AND STANDPIPE	____	NEW
END CAP SEAL	AS-568A-238	PNF MATERIAL
PISTON SEAL, LOW PRESSURE	AS-568A-340	PNF MATERIAL
PISTON SEAL, HIGH PRESSURE	AS-568A-327	BUNA-N-NITRILE (NO MODIFICATION)
PLUG	---	NEW
SEAL, AO-8 SUPPLY	AS-568A-904	PNF MATERIAL
SEAL, AO-8 OUTLET	AS-568A-906	PNF MATERIAL

*DECOTO AIRCRAFT INC. PART NUMBER OR "O" RING SIZE

TABLE 4.2 AO-8 FILL AND BLEED PROCEDURE FOR SYSTEM TESTING

1. APPLY FULL BRAKE PRESSURE

   3000 psi is applied to deboost valve piston

2. OPEN BRAKE ASSEMBLY BLEED VALVE ON EACH BRAKE

   Deboost valve piston moves to the bottom position

3. OPEN AO-8 SUPPLY NEEDLE VALVE

4. PUMP AO-8 INTO DEBOOST VALVE

   AO-8 flows thru the deboost valve and brake assembly purging air from the system

5. CLOSE BRAKE ASSEMBLY BLEED VALVES

6. RELEASE BRAKE PRESSURE

7. PUMP AO-8 INTO THE DEBOOST VALVE UNTIL BRAKE PRESSURE INCREASES TO 200 PSI

   The deboost valve piston will move to the top of the deboost valve

8. CLOSE AO-8 SUPPLY NEEDLE VALVE

9. APPLY FULL BRAKE PRESSURE

10. OPEN ONE BLOCK ASSEMBLY VALVE AND REMOVE 17.0 CUBIC INCHES OF AO-8 FLUID.

11. CLOSE THE BRAKE ASSEMBLY BLEED VALVE

12. RELEASE BRAKE PRESSURE

    The deboost valve piston will be located approximately one inch above the bottom position.

4.2 RECOMMENDED KC-135 BRAKE MODIFICATION

The KC-135 five rotor brake assembly (Federal Stock Number 1630-058-5242) requires no design modification for use in the two-fluid brake hydraulic system. However, since AO-8 hydraulic fluid will be used in the brake, the brake must be assembled with compatible seals. Also, since new KC-135 brake assemblies not exposed to MIL-H-5606 fluid are not available, brakes obtained through the government MILSTRIP system will be reconditioned to comply with the Air Force requirement stated in paragraph 4.8 of the SOW.

Each brake assembly used in the two-fluid brake hydraulic system will be disassembled, cleaned and reassembled using seals compatible with the AO-8 fluid. In addition all moving parts exposed to hydraulic fluid (consisting of the brake piston and piston bushing assemblies) will be replaced with new, untested parts obtained from the brake manufacturer, The Bendix Corporation. The parts replaced or reconditioned and the brake disassembly, cleaning and reassembly procedures are described in the following paragraphs.

4.2.1 REPLACEMENT BRAKE PARTS

The brake parts listed in Table 4.3 will be replaced or cleaned as part of the brake reconditioning procedure. Figure 4.2 is included to help identify the replacement parts and their locations in the brake assembly.

4.2.2 BRAKE DISASSEMBLY

Each KC-135 brake that will be used in the two-fluid brake hydraulic system mockup will be disassembled per the KC-135 Brake Assembly Technical Overhaul Manual (T.O. 4B-1-4-263). The brake piston housing will be removed from the brake stack and disassembled. Brake piston housing disassembly will include the removal of the existing:

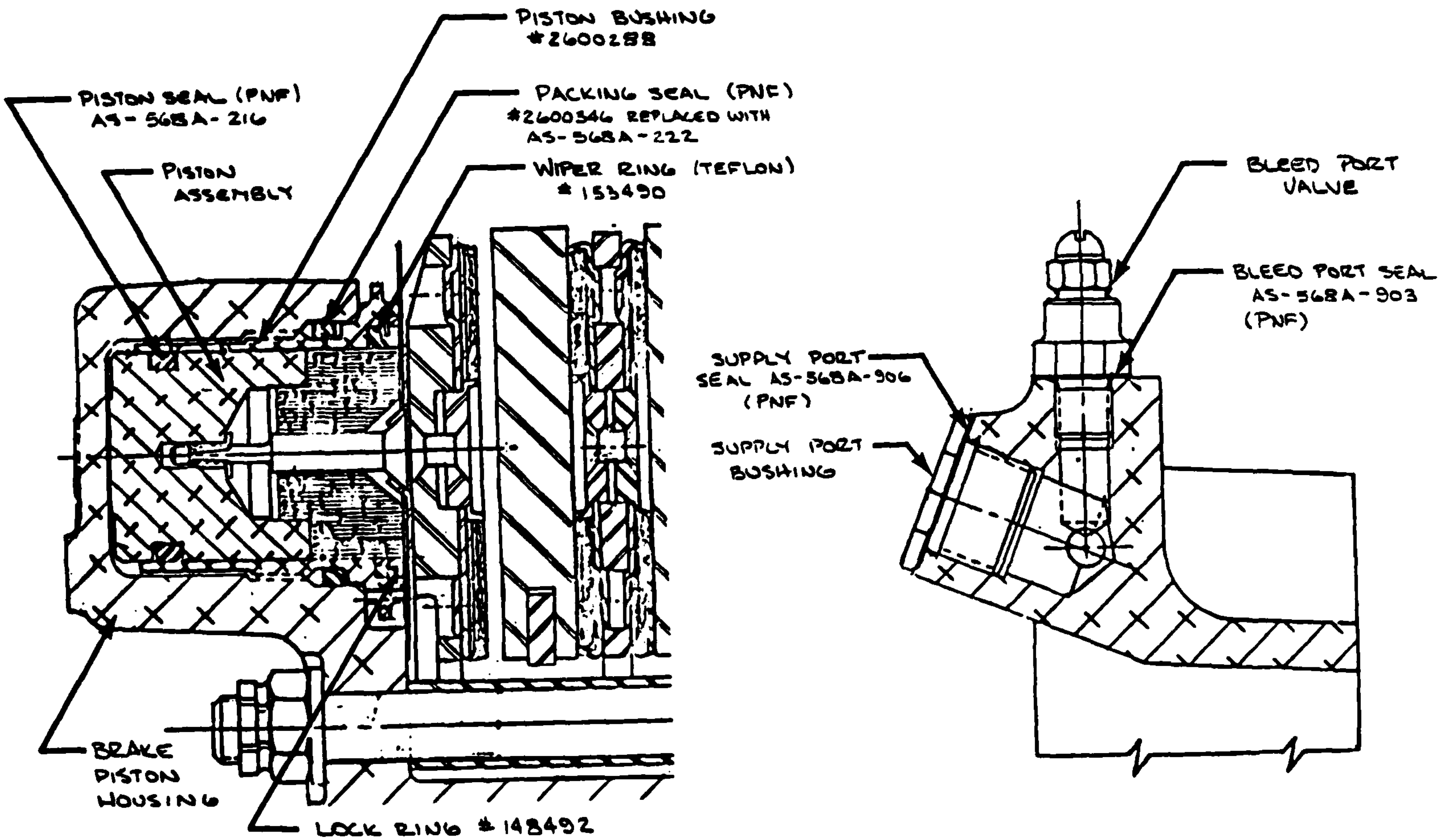
TABLE 4.3 BRAKE RECONDITIONING PARTS

ITEM	QTY*	PART NO.**	STATUS
PISTON BUSHING	8	2600288	NEW
PISTON ASSEMBLY			
PISTON	8	153373	NEW
INSULATOR, PISTON	8	2600384	NEW
PROTECTOR, INSULATOR	8	149293	NEW
SHIELD	8	2600814	NEW
PIN, INSULATOR HOLD DOWN	8	149609	NEW
CLIP, SPLIT TUBULAR	8	149629	NEW
WIPER RING	8	153490	CLEANED
PACKING SEAL***	8	2600346	NEW (PNF)
PISTON SEAL	8	AS-568A-216	NEW (PNF)
LOCKRING	8	148492	CLEANED
DRILL PASSAGE WAY PLUGS	-	-	NEW
BRAKE PISTON HOUSING	1	-	CLEANED
SUPPLY PORT SEAL	1	AS-568A-906	NEW (PNF)
SUPPLY PORT BUSHING	1	-	CLEANED
BLEED PORT VALVE	1	-	CLEANED
BLEED PORT SEAL	1	AS-568A-903	NEW (PNF)

* PER BRAKE ASSEMBLY

** BENDIX PART NUMBER OR SEAL SIZE

*** THE PACKING SEAL IS A SPECIAL MACHINED SEAL, FOR THESE TESTS IT WILL BE REPLACED WITH AN AS-568A-222 PNF SEAL



(A) BRAKE PISTON CROSS SECTION

(B) SUPPLY PORT CROSS SECTION

Figure 4.2 Brake Reconditioning

1) Brake pistons,
2) Brake piston bushings,
3) Brake piston bushing to brake housing seals
4) Drill passageway plugs,
5) Bleed port valve and
6) Fluid supply port fitting.

4.2.3 BRAKE CLEANING PROCEDURE

The brake will be cleaned prior to reassembly as described in Section 4.3.

4.2.4 BRAKE REASSEMBLY

The brake will be reassembled per the procedures detailed in the KC-135 Brake Assembly Technical Overhaul Manual. Reassembly will include the installation of:

1) New brake piston bushings,
2) New brake piston bushing to housing packing seals,
3) New brake piston assemblies,
4) New brake piston seals,
5) New drill passageway plugs,
6) Cleaned fluid supply port bushing with new bushing se[illegible]
7) Cleaned bleed port valve with new seal.

[illegible].3 COMPONENT CLEANING PROCEDURE

[illegible] parts requiring cleaning will be clean[illegible] per the proc[illegible] d be[illegible].

Loose parts such as tubing and unassem[illegible] system com[illegible]

1. Clean with Stoddard Solvent PD-68[illegible]
2. Blow dry and evacuation dry.
3. Rinse with appropriate CTFE fluid [illegible] or [illegible]-2).

For semi-assembled components or parts with deep dead-end passages such as a gage Bourdon tube:

1. Clean with Petroleum Ether.
2. Drain and blow dry with dry nitrogen.
3. Rinse internal passages with appropriate CTFE fluid (AO-8 or AO-2).

This cleaning procedure was developed by the Boeing Materials Technology Laboratory and the Air Force Materials Laboratory during the Air Force sponsored "Fire Resistant Aircraft Hydraulic Systems" contract, F33615-76-C-2064.

SECTION V

COMPONENT AND SYSTEM PERFORMANCE TESTING

5.1 COMPONENT AND SYSTEM PERFORMANCE TEST PLANS

Detailed component performance and system performance test plans were submitted to the Air Force for approval on September 24, 1980. The test plans described the tests necessary to define (1) the component performance of the modified KC-135 deboost valve and brake and (2) the operational and braking performance of the two-fluid brake system. The reader is referred to these plans for a detailed description of the testing. However, a brief discussion of the test objectives and approach is presented here as an overview of the planned effort.

5.2 COMPONENT PERFORMANCE TESTS

Each KC-135 deboost valve and KC-135 brake modified for use in the AO-8 two-fluid brake hydraulic system mockup will be tested to (1) assure that each component meets the production part functional performance acceptance test requirements prior to its installation in the mockup and (2) determine its dynamic response characteristics. The component performance tests which will be conducted on the deboost valve and brake are listed in Table 5.1. A detailed description of each test can be found in the Component Performance Test Plan submitted to the Air Force. The results of these tests will be compared to similar tests run on unmodified deboost valve and brake components to determine the effects of the two-fluid modifications.

The functional performance testing includes tests for proof pressure, static and dynamic leakage and piston friction. These tests will be performed at ambient, -65 degrees F. and 160 degrees F. to cover the temperature range of normal brake system operation. The functional performance tests will be conducted on an individual component basis before installation in the brake hydraulic system mockup.

TABLE 5.1 COMPONENT PERFORMANCE TESTS

DEBOOST VALVE COMPONENT PERFORMANCE TESTS

EXAMINATION OF PERFORMANCE TESTS

SEAL BREAK IN

PROOF PRESSURE AND STATIC LEAKAGE

DYNAMIC LEAKAGE

SEAL FRICTION

BRAKE COMPONENT PERFORMANCE TESTS

EXAMINATION OF PRODUCT

SEAL BREAK IN

PROOF PRESSURE AND STATIC LEAKAGE

DYNAMIC LEAKAGE

The component dynamic response tests are designed to define the dynamic characteristics of the modified deboost valve and brake at typical operating conditions. Frequency response tests conducted at ambient, -65 degrees F. and 160 degrees F. will be performed to determine the dynamic response characteristics (gain and phase angle) of each component. The individual component dynamic response tests will be performed with the modified component installed in the brake hydraulic system mockup. This approach is the correct method of determining component response since the characteristics of an individual component change with the hydraulic load. For example, the natural frequency of an antiskid valve when determined in a blocked port test is above 100 Hertz. However, when a hydraulic load (such as a brake assembly) is connected to the output port of the same antiskid valve the natural frequency of the valve is approximately 30 Hertz. Since component performance is dependent upon system configuration it is essential that the dynamic response of each component be determined with the component in its proper system configuration.

5.3 SYSTEM PERFORMANCE TESTS

A series of system tests will be conducted to determine (1) the overall system operation (dynamic response) and (2) the stopping performance of the KC-135 two-fluid brake system. The system performance tests which will be conducted are listed in Table 5.2. A detailed description of each test can be found in the System Performance Test Plan submitted to the Air Force. The results of these tests will be compared with similar tests conducted with an unmodified KC-135 brake hydraulic system to determine the effects of the AO-8 two-fluid brake system configuration.

The system operational tests, which include frequency response and step response tests, will be performed to define the dynamic characteristics (gain and phase angle) of the brake hydraulic system and select components within the system. During these tests a sinusoidal or step electronic control input signal will be made to the antiskid valve and compared to the pressure response at several locations within the system. The tests will be conducted at ambient, -65 degrees F. and 160 degrees F. to cover the temperature range of normal brake system operation.

TABLE 5.2 SYSTEM PERFORMANCE TESTS

SYSTEM OPERATIONAL CHARACTERISTICS TESTS

TEST 1. FREQUENCY RESPONSE

TEST 2. STEP RESPONSE

TEST 3. STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE

TEST 4. STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME

STOPPING PERFORMANCE TESTS

TEST 5. CONSTANT FRICTION RUNWAY

TEST 6. WET RUNWAY

TEST 7. STEP FRICTION

TEST 8. LANDING GEAR SYSTEM STABILITY

Static antiskid valve current versus brake pressure and static brake pressure versus brake volume tests will be performed to determine the characteristics of the antiskid valve and brake components respectively. These tests are for information only and will be used to determine whether or not these components meet the manufacturer's specifications. The tests will be performed at ambient temperature only since these component characteristics are not temperature dependent.

The objective of the stopping performance tests are to determine the impact of the two-fluid brake hydraulic system on braking performance and define the response of the brake system to antiskid signals. These tests will provide a functional checkout of the operation and performance of the overall brake system. A hybrid computer simulation of the KC-135 aircraft and landing gear systems, along with the KC-135 two-fluid brake hydraulic system mockup and an active antiskid control system will be used to determine the system performance. The stopping distance of the aircraft and the system response to antiskid signals will be determined at a variety of environmental conditions (i.e., runway friction; dry, wet, icy and icy patches) and at ambient temperature, -65 degrees F. and 160 degrees F. to covered the range of normal system operating conditions. The stopping distance and antiskid signal response of the two-fluid system will be compared to similar data generated with the normal KC-135 brake system.

In addition, a landing gear stability study will be performed to determined the impact of the two-fluid brake system configuration on gear walk and antiskid system stability.

5.4 HYDRAULIC FLUID SAMPLES

Samples of the AO-8 hydraulic fluid (8-ounce size) will be taken periodically and provided to AFWAL/MLBT throughout the system tests. The samples will be taken after 0, 2 and 5 hours of testing and at 5 hour increments thereafter (not including temperature soak time).

APPENDIX A-1

HSFR PROGRAM MODIFICATION

A.1 PROGRAM MODIFICATIONS

The government owned Hydraulic System Frequency Response (HSFR) computer program was modified to permit dynamic analysis of the two-fluid brake hydraulic system. The changes made to the program were:

(1) Definition of fluid properties (bulk modulus, density and viscosity) as subscripted variables

(2) Inclusion of AO-8 and AO-2 fluid properties in the program

(3) Addition of a subroutine to define the flow and pressure relationship associated with the deboost valve piston and

(4) Modification of the accumulator subroutine to model the brake.

Each program change is discussed below. No attempt has been made to discuss the implementation of the changes; however, the purpose of the change is discussed and equations (where necessary) are supplied.

A.2 SUBSCRIPTED FLUID PROPERTIES

The variable names of the fluid properties (bulk modulus, density and viscosity) found within each component subroutine have been changed from a simple variable to a subscripted variable. In addition a fluid specification variable was added to each subroutine so the user can specify the type of fluid found within a particular component (i.e., whether MIL-H-5606 or AO-8 is used in a hydraulic line, accumulator, pump etc.). These changes were required to permit evaluation of a system containing two (or more) types of hydraulic fluid.

The fluid properties at a particular operating condition (i.e., temperature and pressure) are calculated with the FLUID subroutine. A FORTRAN do loop was added along with subscripts on the temperature and pressure variables (associated with each fluid) to calculate the properties of each hydraulic fluid.

A.3 AO-8 AND AO-2 FLUID PROPERTIES

The AO-8 and AO-2 fluid properties listed in Table A.1 were inserted in the HSFR fluid properties subroutine, FLUID.

A.4 DEBOOST VALVE PISTON MODEL

A FORTRAN subroutine modelling the flow and pressure output of the deboost valve piston was added to HSFR. The effects of different input and output piston areas, piston mass and damping are included in the model. The subroutine calculates a 2 x 2 matrix relating input flow and pressure to output flow and pressure. The matrix relationship for the deboost valve piston flow and pressure is.

$$\begin{bmatrix} \frac{A_2}{A_1} & 0 \\ -\frac{mS}{A_1A_2} - \frac{b}{A_1A_2} & \frac{A_1}{A_2} \end{bmatrix} \times \begin{bmatrix} Q_1 \\ P_1 \end{bmatrix} = \begin{bmatrix} Q_2 \\ P_2 \end{bmatrix}$$

TABLE A.1 FLUID PROPERTIES

FLUID PROPERTY	TEMPERATURE (DEGREES F)	MIL-H-5606*	CTFE FLUIDS HALOCARBON AO-8**	AO-2**
ADIABATIC	-65	13.47	13.38	13.38
TANGENT BULK	-40	3.25	3.17	3.17
MODULUS X 10^{-5}	0	2.9	2.82	2.82
	50	2.48	2.40	2.40
PSI	100	2.08	2.00	2.00
	150	1.73	1.65	1.65
	200	1.42	1.34	1.34
	250	1.19	1.11	1.11
	300	.98	.90	.90
KINEMATIC	-65	1993.5	2800.0	1100.0
VISCOSITY	-40	482.3	540.0	200.0
	0	134.4	82.0	30.0
	50	34.85	18.7	7.5
CENTISTOKES	100	14.47	7.3	3.1
	150	7.46	3.75	1.72
	200	4.58	2.35	1.08
	250	3.19	1.66	.74
	300	2.39	1.26	.5
VISCOSITY PRESSURE CORRECTION COEFFICIENT	--	.335	.3929	.445
DENSITY X 10^5	-65	8.57	18.70	18.70
LB-SEC^2/IN^4	275	7.63	15.61	15.61

* Data from Air Force HSFR Computer program

** Viscosity data obtained from AFWAL/MLBT. Bulk Modulus and density data obtained from Fire Resistant Aircraft Hydraulic Systems, AFWAL TR-80-2112.

Where A_1 = Piston area, input side - in**2

A_2 = Piston area, output side - in**2

b = Piston damping - lb *sec/in

m = Piston mass - lb*sec**2/in

P_1 = Pressure, input - lb/in**2

P_2 = Pressure, output - lb/in**2

Q_1 = Flow, input - in**3/sec

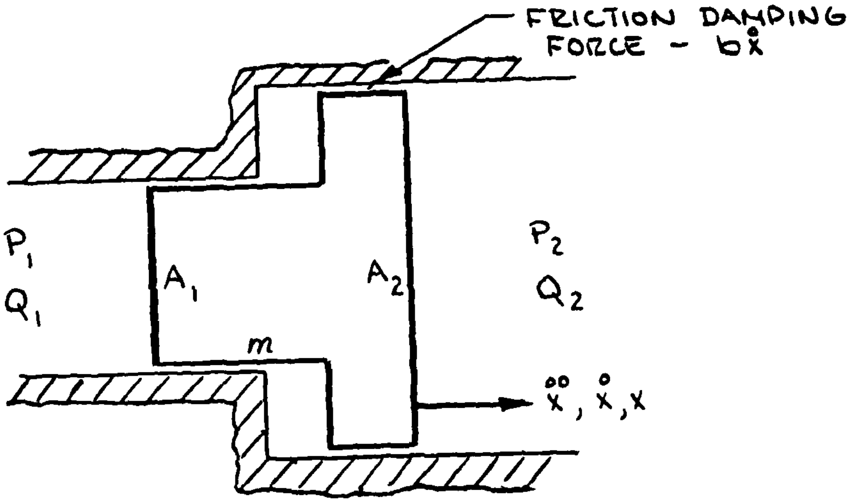
Q_2 = Flow, output - in**3/sec

S = Laplace operator - 1/sec

The piston model is shown in Figure A.1

A.5 BRAKE MODEL

The standard HSFR accumulator subroutine was modified to model the brake. The stiffness term associated with the accumulator air spring was changed to a linear stiffness term. This change was made so that brake stiffness could be varied independent of the accumulator (brake) pressure.



FORCE EQUATION

$$m\ddot{x} = P_1 A_1 - P_2 A_2 - b\dot{x}$$

FLOW EQUATIONS

$$Q_1 = A_1 \dot{x}$$

$$Q_2 = A_2 \dot{x}$$

Figure A.1 Deboost Valve Piston Model

APPENDIX B

COMPONENT MODIFICATIONS FOR TESTING

B.1 PNF O-RINGS

The PNF seal compound was supplied by the Firestone Tire and Rubber Company, Akron, Ohio. The formulation of the PNF compound (Firestone compound PNF-280-001R, 80 Durometer) was specified by AFWAL/MLBT.

The PNF-280-001R O-ring seals were molded by Lord Kinematics, Shelton, Connecticut.

B.2 DEBOOST VALVE MODIFICATIONS

The deboost valve modifications which were made for the laboratory tests are described in Appendix A. The detail mechanical drawings used to fabricate parts for the modification are given in Figures B-1, B-2 and B-3.

B.3 BRAKE MODIFICATIONS

The brake modifications which were made for the laboratory tests are described in Appendix A. Details of the modifications and brake reconditioning are described below.

B.3.1 BRAKE PISTON SEALS

During the Interim Report Presentation at WPAFB, November 13, 1980, it was learned that the original brake piston O-ring seal (MS-28775-216) called out on the Bendix KC-135 brake assembly drawing is being replaced during brake overhaul with a T-seal. Since PNF O-rings were to be used in the brakes exposed to CTFE hydraulic fluid, it was decided with the concurrence of the Air Force Project Engineer, Mr. Bruce Campbell that the T-seals in the standard fluid brakes would be replaced with the original MS 28775-216 O-rings. Use of O-rings in both the CTFE and MIL-H-5606 fluid brakes eliminates the possibility of performance differences caused by different seal

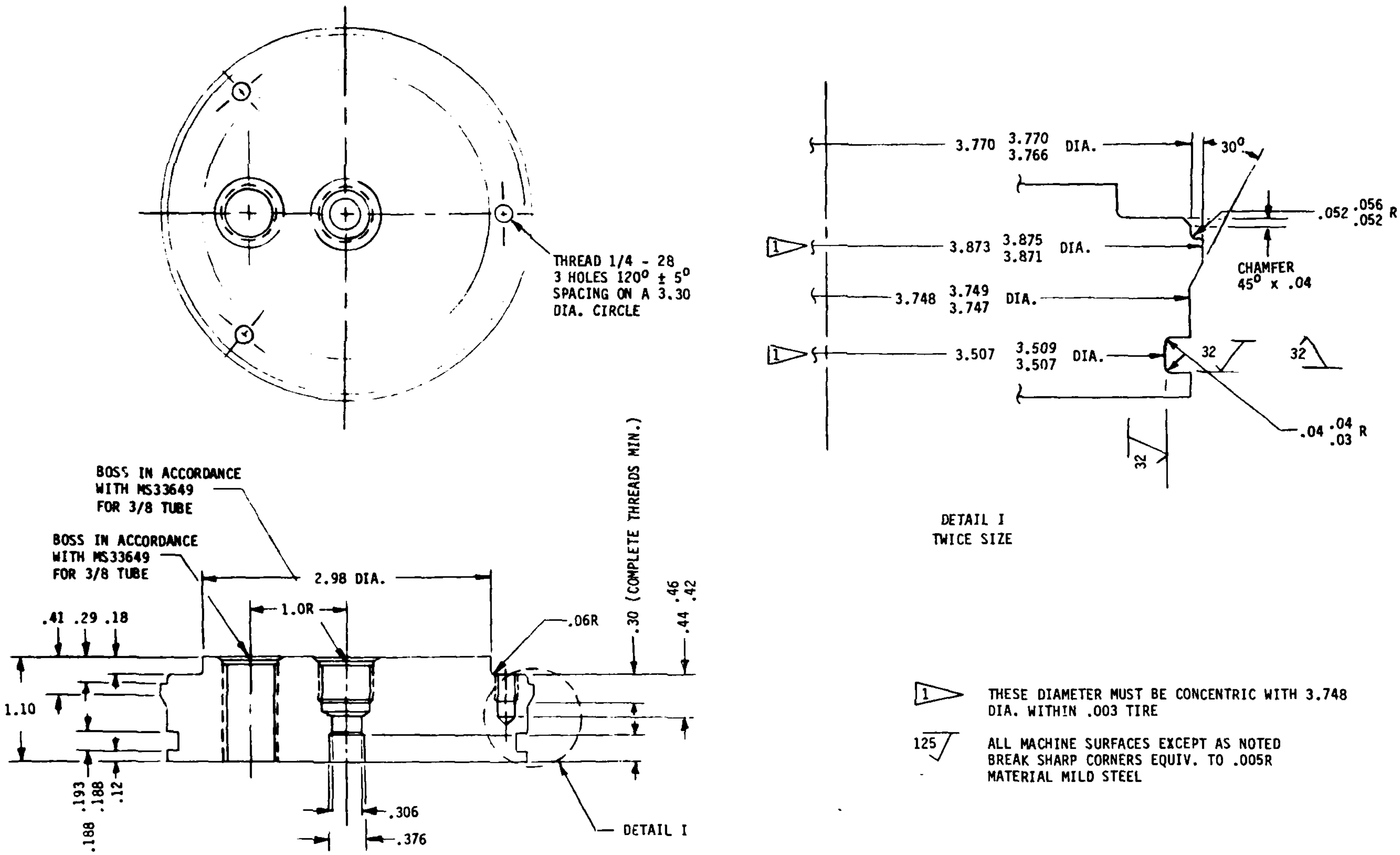


Figure B-1 Endcap Modification

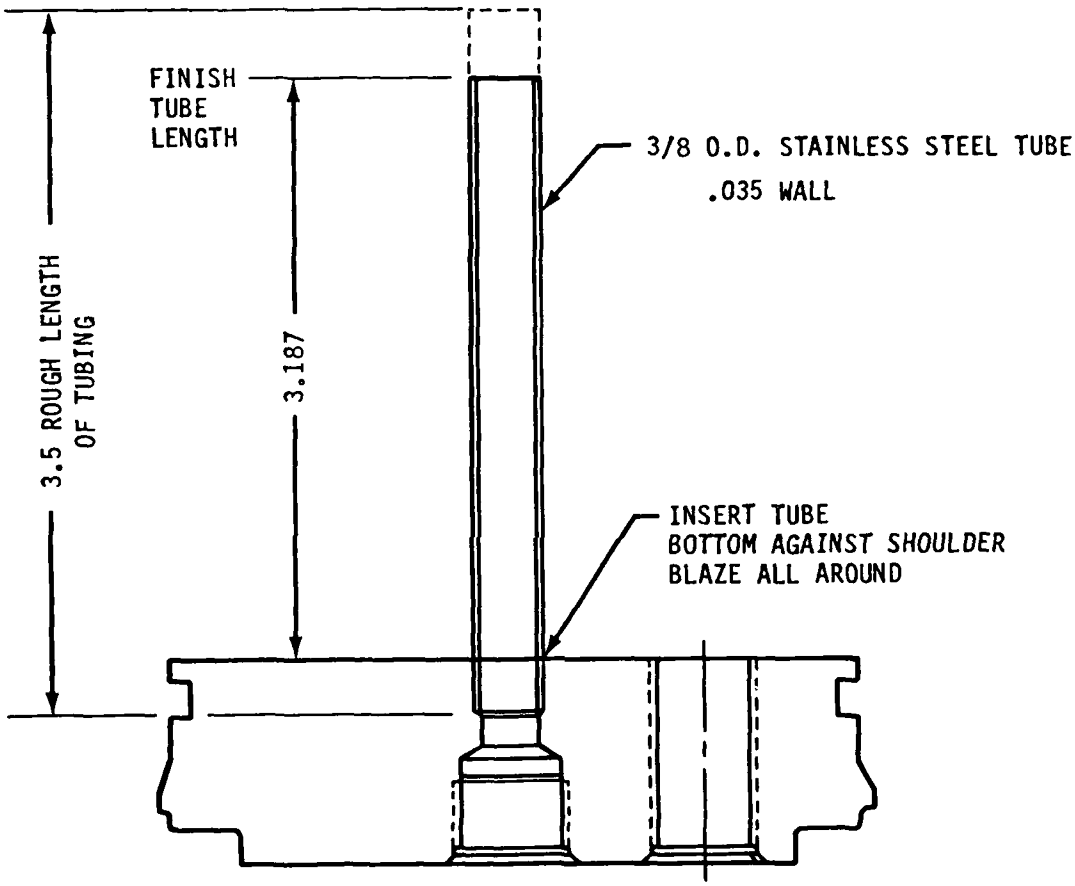


Figure B-2 Endcap and Standpipe

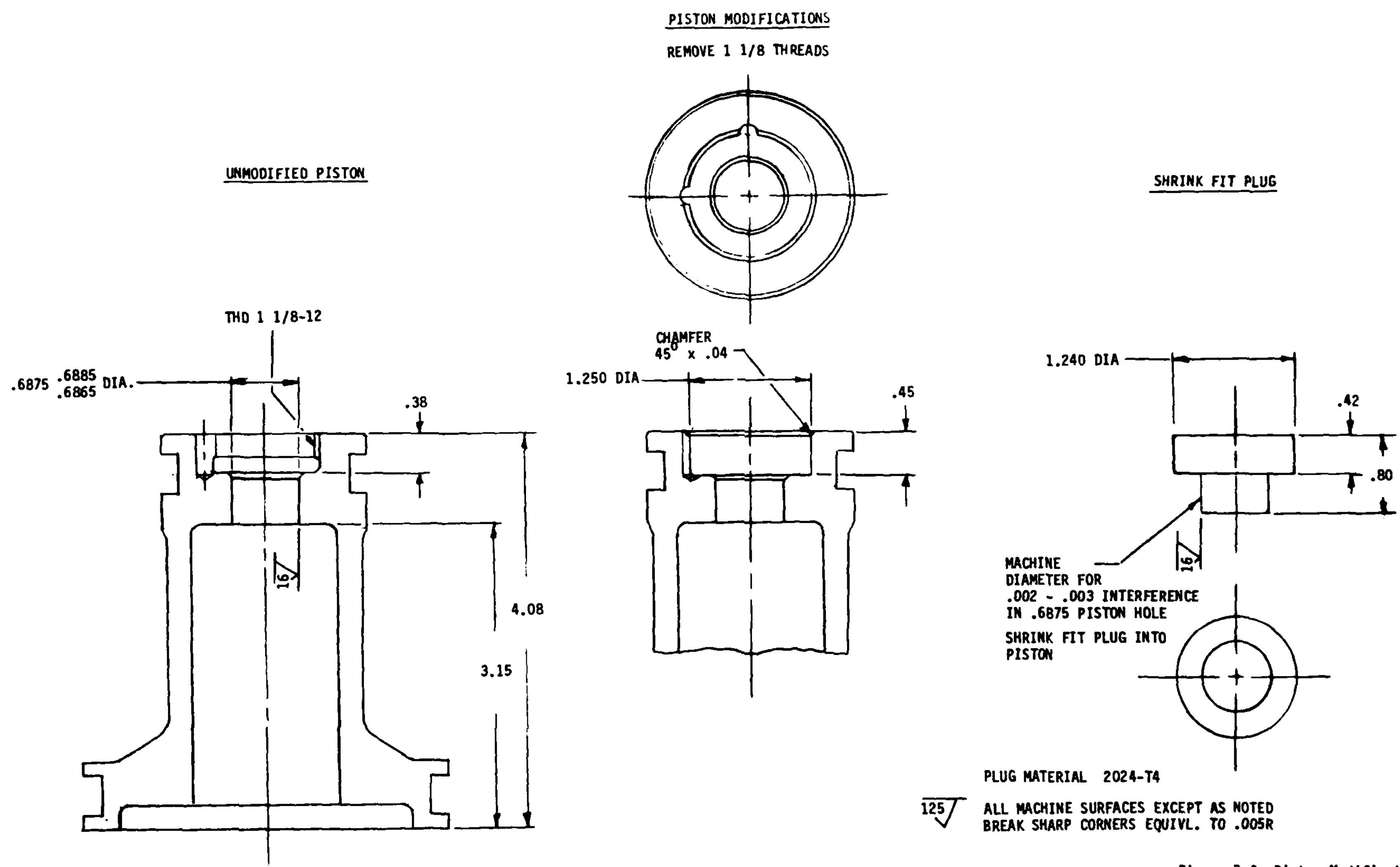


Figure B-3 Piston Modification

configurations. The static bushing seal (Bendix part number 2600346, Figure 4.2, Appendix A) used in the KC-135 brake is a special milled packing. This packing was replaced with a AS-568A-222 PNF O-ring in the brakes exposed to CTFE fluid.

B.3.2 BRAKE RECONDITIONING

The two KC-135 5-rotor brakes (obtained through the government MILSTRIP system) which were exposed to the CTFE hydraulic fluid were reconditioned prior to laboratory testing.

Each brake assembly was disassembled, cleaned and reassembled using PNF O-ring seals.

Disassembly of each brake included removal of:

(1) the backing plate, rotors and stators from the brake hydraulic housing,
(2) the pressure plate, and retractor spring assemblies from the brake hydraulic housing,
(3) the brake piston and piston bushing assemblies from the brake hydraulic housing,
(4) the piston bushing to brake housing seals and
(5) the pressure port bushing and the bleed valve from the brake hydraulic housing.

The drill passageway plugs in the brake housing were not removed for reconditioning. It was found during the disassembly that the plugs were not located in deep dead end drilled passages as anticipated, but were exposed (in the brake piston cavity) and easily cleaned. Mr. Bruce Campbell, Air Force Project Engineer, was consulted and agreed with the decision not to remove the plugs.

The brake housing, supply port bushing, retractor spring assemblies, teflon wiper rings and lock rings were cleaned as described below:

(1) All parts were vapor degreased using a hot bath of perchloroethylene. Each part was c' aned with a high pressure fluid spray. Special attention was given to cleaning the drilled passageways and ports in the brake housing.
(2) All parts (including the replacement brake pistons and piston bushing in addition to the parts listed above) were cleaned with Stoddard Solvent PD-680.
(3) Each part was drained and blown dry.
(4) Surfaces and parts exposed to the CTFE fluid during normal operation were coated with the CTFE fluid before reassembling.

Each brake was reassembled per the procedures detailed in the KC-135 Brake Technical Overhaul Manual (T.O. 4B1-4-263). Reassembly included:

(1) Installation of new piston bushings and PNF O-ring bushing seals in the cleaned brake housing.
(2) Installation of the cleaned teflon wiper rings and lock rings in the new piston bushings.
(3) Installation of new brake pistons with PNF O-ring piston seals in the piston bushings.
(4) Assembly of the pressure plate retractor spring mechanisms in the brake hydraulic housing.
(5) Assembly of the braking plate, rotors, stators and brake hydraulic housing.
(6) Installation of the cleaned supply port bushing and PNF O-ring bushing seal in the brake housing.

APPENDIX C

LABORATORY TEST PLAN

The Component Performance Test Plan and System Performance Test Plan document which was submitted to and approved by the Air Force is reprinted in the following pages.

FIREPROOF BRAKE HYDRAULIC SYSTEM

COMPONENT PERFORMANCE TEST PLAN

AND

SYSTEM PERFORMANCE TEST PLAN

(REVISED)

CONTRACT F33615-80-C-2026
PROJECT 3145
CDRL SEQUENCE NO. 4

SEPTEMBER 1980
REVISED NOVEMBER 1980

SUBMITTED FOR APPROVAL OF:

AIR FORCE AERO PROPULSION LABORATORY
ATTN: BRUCE CAMPBELL, AFWAL/POOS-1

PRINCIPAL INVESTIGATOR Steven M Warren PROGRAM MANAGER J R Kilner
S. M. WARREN J. R. KILNER

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SECTION I

INTRODUCTION

The component performance and system performance tests outlined herein meet the requirements specified in Section 6, Paragraphs 4.7, 4.8, and 4.9 of contract F33615-80-C-2026.

1.1 COMPONENT PERFORMANCE TESTING

Each KC-135 deboost valve and KC-135 brake assembly modified for use in the AO-8 fluid hydraulic brake system mockup will be tested to assure that the performance of each component meets the production part acceptance test requirements for proof pressure, leakage and piston friction prior to installation of the part in the mockup. The tests, test procedures, requirements, etc., to which each modified deboost valve or brake assembly will be subjected are outlined in Sections II and III.

1.2 SYSTEM PERFORMANCE TESTING

A series of system tests will be conducted to determine the operational characteristics and braking performance of the two-fluid hydraulic brake system. The operational characteristic tests will determine the hydraulic component and system response to antiskid signals. These will include frequency and step response tests conducted to determine the dynamic characteristics of the two-fluid interface unit (modified deboost valve) and the overall hydraulic brake system. These tests are described in Section IV.

The braking performance tests utilizing a hybrid computer simulation of the KC-135 aircraft and landing gear, the brake system mockup, and an active antiskid control system, will determine the stopping performance of the two-fluid brake system. The system performance tests are detailed in Section IV.

The system performance tests will be performed with both the two-fluid system and a single-fluid hydraulic brake system (i.e., unmodified KC-135 hydraulic brake system). The single-fluid system will be used to establish baseline performance data. To evaluate the performance of the two-fluid hydraulic brake system, baseline data will be compared with the two-fluid system data.

1.3 TEST FACILITY

All tests will be performed in the Mechanical Systems Laboratory at the Boeing Developmental Center, Seattle, Washington. The equipment and test setups required for each component and system performance test are described in the section detailing the respective tests.

SECTION II

DEBOOST VALVE, COMPONENT PERFORMANCE TESTING

The KC-135 deboost valve modified for use in the two-fluid hydraulic brake system mockup will be tested to ensure that the modified valve meets the acceptance test performance requirements of new unmodified production units. The tests and procedures detailed below are based upon the manufacturer's (The Decoto Aircraft Company, Yakima, Washington) recommended test procedure (see Appendix C-1).

2.1 DISCUSSION OF PROBLEMS AND TEST OBJECTIVE

Seal friction and leakage are the primary factors which can effect the performance of the deboost valve in the two-fluid hydraulic brake system. Seal leakage presents service and material contamination problems. Excessive leakage requires fluid replacement (i.e., servicing) and may also lead to fluid mixing and exposure of MIL-H-5606 seals to AO-8 fluid. This exposure and fluid mixing can result in the formation of a precipitate which may restrict free deboost valve piston motion and ultimately cause a loss in braking performance. Excessive seal friction may also reduce free piston motion and cause a loss in braking performance.

The objective of the tests described below is to assure that the leakage and friction problems have been solved prior to the installation and testing of the modified deboost valve in the two-fluid brake hydraulic system and that the valve meets the performance requirements of new unmodified production units.

2.2 DEBOOST VALVE TESTS

The modified KC-135 deboost valve will be subjected to the following functional tests prior to its installation and testing in the two-fluid brake hydraulic system mockup.

1. Examination of Product
2. Seal Break In

3. Proof Pressure and Static Leakage
4. Dynamic Leakage
5. Seal Friction

These tests will be conducted using MIL-H-5606 hydraulic fluid and MIL-H-5606 compatible seals on the high pressure side of the deboost valve (Figure 2.1) and AO-8 fluid and PNF seals on the low pressure side (brake side).

The Examination of Product, Seal Break In, leakage and friction tests will be performed on the modified KC-135 deboost valve at ambient temperature. The dynamic leakage and friction tests will also be performed on the modified valve at -65 degrees F and 160 degrees F. In addition, the dynamic leakage and seal friction test will be performed on an unmodified deboost valve (single fluid system with MIL-H-5606 fluid) at the same temperature conditions as the modified valve. Detailed explanations of each test, test objective, the test sequence, recorded data, instrumentation, etc., are given in the following paragraphs.

2.3 TEST SETUP

The modified deboost valve will be installed in the test setup shown schematically in Figure 2.2 All deboost valve component performance testing will be accomplished with this setup. The number and approximate location of all necessary test instrumentation are shown or described in Figure 2.2.

The test setup will be placed in an environmental chamber for those tests conducted at temperatures other than ambient room temperature.

2.4 TEST EQUIPMENT

The equipment and instrumentation used during the deboost valve tests are listed in Table 2.1. The range, accuracy and resolution required for each piece of equipment used for data acquisition are listed in the table.

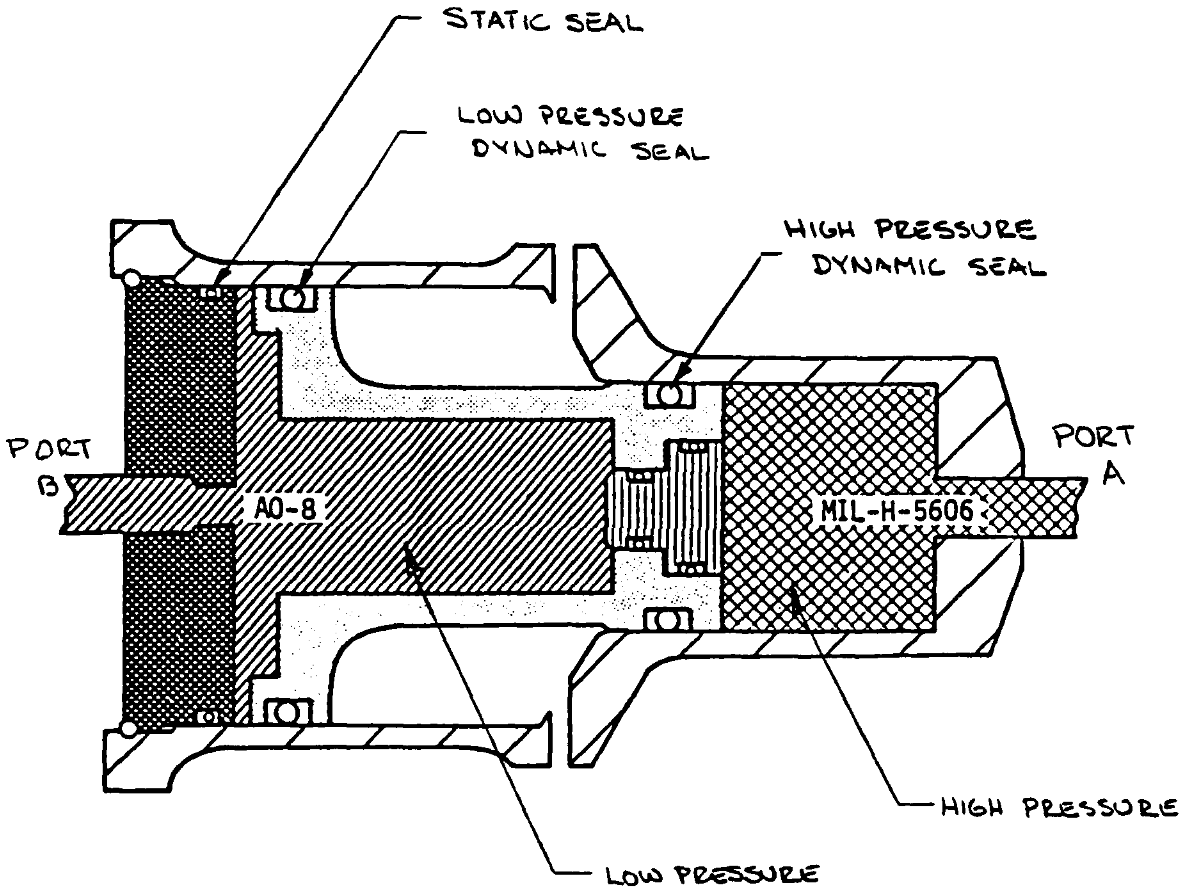

Figure 2.1 Deboost Valve

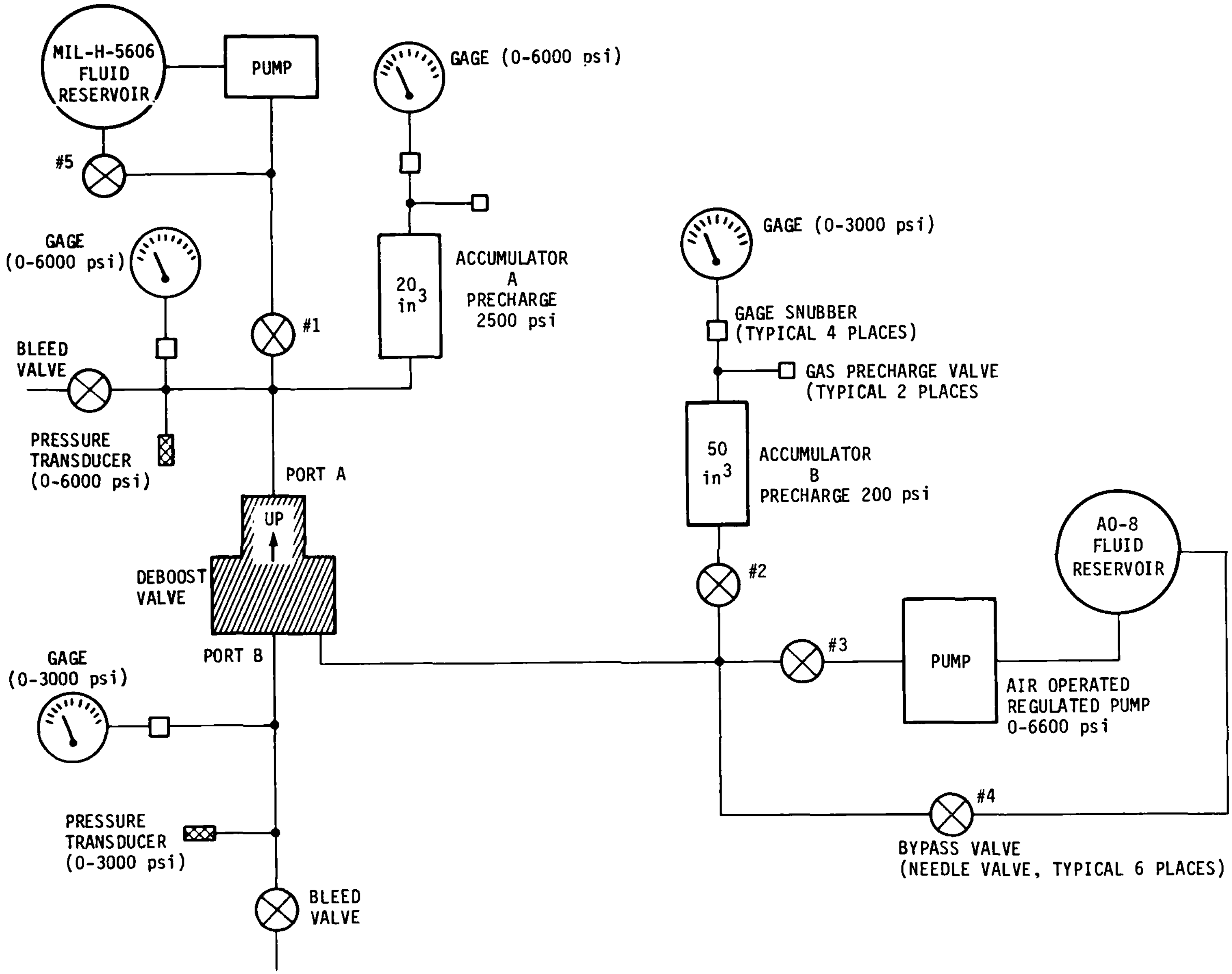

Figure 2.2 Test Setup - Deboost Valve, Component Performance Testing

TABLE 2.1 TEST EQUIPMENT - DEBOOST VALVE, COMPONENT PERFORMANCE TESTING

ITEM	QTY.	RANGE	ACCURACY	RESOLUTION	USE	COMMENT
Pressure Gage	2	0 - 3000 psi	5 psi	10 psi	Accumulator and Deboost Valve Pressure	Indication Only
Pressure Gage	2	0 - 6000 psi	5 psi	10 psi	Accumulator and Deboost Valve Pressure	Indication Only
Pressure Transducer	1	0 - 6000 psi	2 psi	1 psi	Deboost Valve Pressure, High Pressure Side	
Pressure Transducer	1	0 - 3000 psi	2 psi	1 psi	Deboost Valve Pressure, Low Pressure Side	
Thermocouple	1	-100 to 212 F	2°F	1°F	Environmental Temperature	
Environmental Chamber/ Temperature Capability	1	-100 to 212 F	5°F	—	Temperature Control	
AO-8 Fluid Regulated Pressure Supply Source	1	0 - 6600 psi	—	—	Fluid/Pressure Source	
MIL-H-5606 Fluid Regulated Pressure Supply Source	1	0 - 6600 psi	—	—	Fluid/Pressure Source	
Brush Chart Recorder (8 Channel)	1	—	—	—	Time History Data	

2.5 TEST DESCRIPTIONS

Reference to the test setup schematic (Figure 2.2) will provide additional insight into the test descriptions given below.

2.5.1 EXAMINATION OF PRODUCT, TEST 1

The modification and assembly of the KC-135 deboost valve which will be subjected to the AO-8 hydrualic fluid will be supervised and observed by the test engineer to assure that the modified deboost valve conforms to the manufacturer's specifications.

2.5.2 SEAL BREAK IN, TEST 2

Prior to performance testing the deboost valve will be subjected to a break in period of 200 cycles of the application and release of pressure per the test procedure detailed in the dynamic leakage test (Section 2.5.4). The objective of this break in period is to assure proper seating of dynamic seals and wear any residual mold release agent or lubricant from the seals prior to the start of testing.

LEAKAGE TESTS

Static and dynamic leakage tests will be performed to assure that the application of AO-8 hydraulic fluid and the PNF seals in the deboost valve meets the manufacturer seal leakage performance requirements.

2.5.3 PROOF PRESSURE AND STATIC LEAKAGE TESTS

Two static leakage tests will be performed to determine the fluid leakage past the high pressure piston seal and the low pressure piston seal (Figure 2.1) under static conditions at proof pressure.

2.5.3.1 PROOF PRESSURE AND HIGH PRESSURE SEAL STATIC LEAKAGE, TEST 3

Test Objective

Measure the fluid leakage past the high pressure deboost valve piston seal under static conditions at proof pressure.

Test Procedure

Depressurize the deboost valve at Port B (i.e., 0.0 psi*) by opening valve #4. Then, apply pressure to Port A to bottom the piston at the large end (as observed through the vent hole). With the piston bottomed apply 4500 psi to Port A and hold for 2 minutes then reduce pressure to 5 psi and hold for 2 minutes.

Performance Requirement

There shall be less than one drop of fluid leakage as observed through the vent hole.

2.5.3.2 PROOF PRESSURE AND LOW PRESSURE SEAL STATIC LEAKAGE, TEST 4

Test Objective

Measure the fluid leakage past the low pressure deboost valve dynamic piston seal under static conditions at proof pressure.

Test Procedure

Depressurize Port A (i.e., 0.0 psi) by opening valves #1 and #5 and apply pressure to Port B to bottom the piston at the small end (as observed through the vent hole). With the piston bottomed apply 1445 psi to Port B and hold for 2 minutes then reduce pressure to 5 psi and hold for 2 minutes.

* All pressures are referenced to ambient pressure (i.e., gauge).

Performance Requirement

There shall be less than one drop of fluid leakage as observed through the vent hole.

2.5.4 DYNAMIC LEAKAGE, TEST 5

Test Objective

Measure the fluid leakage past the dynamic seals of the deboost valve under dynamic pressure conditions.

Test Procedure

Depressurize Port B (i.e., 0.0 psi) by opening valve #4 and apply pressure to Port A to bottom the piston at the large end. With the piston bottomed apply 600 psi to Port A. Then, close the MIL-H-5606 supply valve (valve #1). Cycle pressure at Port B between 0.0 psi and 965 psi 25 times. The deboost valve shall be allowed to stabilize at the test temperature before testing (a minimum temperature soak times of 6 hours will be observed).

Performance Requirement

Piston leakage as observed through the vent hold shall not exceed one drop.

2.5.5 SEAL FRICTION, TEST 6

Test Objective

Measure the seal friction load which must be overcome before piston motion occurs.

Test Procedure

Depressurize Port B (i.e., 0.0 psi) by opening valve #4, and pressurize Port A to bottom the piston at the large end. With the piston bottomed apply 2000 psi to Port A. Close the MIL-H-5606 supply valve (#1), open Accumulator B supply valve (#2) and pressurize Port B to 965 psi (the pressure at Port A should be approximately 3000 psi). Close Accumulator B supply valve (#2) and the AO-8 supply valve (#3). Slowly reduce pressure at Port B to 800 psi by opening the AO-8 bypass valve (#4). Close the bypass valve and record the pressure at Ports A (P_{A3}) and B (P_{B3}). Slowly increase pressure at Port B by opening Accumulator B supply valve (#2) until the piston starts to move (or the pressure at Port A starts to increase), record the pressure at Port A (P_{A5}) and Port B (P_{B5}). The seal friction force F can then be calculated using the following equation.

$$F = (P_{B5} - P_{B3})\ 11.0270 - (P_{A5} - P_{A3})\ 3.5332$$

Performance Requirement

No known deboost valve seal friction requirement or statistical data for comparison exists, thus no-performance requirement has been set.

This test will be performed on both the modified and unmodified deboost valves. The results of these tests will be recorded, compared and used in later tests to help explain possible performance differences.

2.6 TEST CONDITIONS

The modified KC-135 deboost valve will be tested at ambient laboratory conditions as described in Sections 2.2 thru 2.5. The modified valve will also be subjected to the Dynamic Leakage test (see Section 2.5.4) and the Seal Friction test (see Section 2.5.5) at temperatures of -65 degrees F and 160 degrees F. In addition to these tests, an unmodified KC-135 deboost valve will be subjected to the same Dynamic Leakage and Seal Friction tests as for the modified valve. The conditions associated with each test are given in the Test Outline, Table 2.2.

2.7 TEST SCHEDULE

The deboost valve tests shall be performed in the sequence listed in the Test Outline, Table 2.2. The data to be recorded during each test and the number of runs per test are given in Table 2.2.

2.8 FLUID SAMPLES

A sample of the AO-8 hydraulic fluid supplied to Boeing by AFWAL/MLBT will be analyzed at Boeing to determine the total acid number and kinematic viscosity at ambient temperature. In addition, samples of the AO-8 hydraulic fluid (8-ounce size) will be taken at the beginning and end of the deboost valve tests and supplied to AFWAL/MLBT.

TABLE 2.2 TEST OUTLINE - DEBOOST VALVE, COMPONENT PERFORMANCE TESTING

TEST	TEST CONDITIONS PRESS. -psi-	TEST CONDITIONS TEMP. -°F-	REPETITIONS	RECORDED DATA	COMMENT
1. Examination of Product	---	---	---	Comments as Necessary	
2. Seal Break In	0 - 300 0 - 600 0 - 965	Ambient	50 Cycles 50 Cycles 100 Cycles	Comments as Necessary	
3. Proof Pressure and High Pressure Seal Static Leakage	4500	Ambient	1	Pressure Fluid Leakage	
4. Proof Pressure and Low Pressure Seal Static Leakage	1445	Ambient	1	Pressure Fluid Leakage	
5. Dynamic Leakage Test				Pressure Temperature Fluid Leakage Cycles	
5a Ambient	0 - 965	Ambient	1		Test performed on both a modified and unmodified valve
5b -65°F	0 - 965	-65	1		
5c 160°F	0 - 965	160	1		
6. Seal Friction				Pressure Temperature	
6a Ambient		Ambient	3		
6b -65°F		-65	3		
6c 160°F		160	3		

SECTION III

BRAKE, COMPONENT PERFORMANCE TESTING

Each KC-135 brake modified for use in the two-fluid hydraulic brake system mockup will undergo functional testing prior to its installation in the mockup to ensure that the brake meets the performance requirements of new unmodified production brake units. The brake will be tested per the Aircraft Wheel and Brake Assembly Military Specification MIL-W-5013H. MIL-W-5013H is currently used by the brake manufacturer, the Bendix Corporation, South Bend, Indiana, for acceptance testing of production KC-135 brakes.

3.1 DISCUSSION OF PROBLEMS AND TEST OBJECTIVE

The production KC-135 brakes which will be exposed to AO-8 fire resistant hydraulic fluid must be modified by replacing all static and dynamic seals with PNF seals which are compatiable with the AO-8 fluid. There are two potential problems, excessive leakage and excessive seal friction, which may occur as a result of this modification. Excessive leakage may lead to servicing problems and a reduction in the torque capability of the brake if fluid contaminates the brake stack, while excessive seal friction may not allow the brake piston to fully retract resulting in a dragging brake, accelerated brake wear and reduced brake system performance.

It is the objective of the functional tests described below to ensure the such problems as leakage and friction are solved prior to the installation and testing of the modified brake in the two-fluid hydraulic brake system and that the brake meets the performance requirements of new unmodified production units.

3.2 BRAKE ASSEMBLY TESTS

Each KC-135 brake modified for use in the two-fluid hydraulic brake system will be tested as specified in Quality Conformance Tests, Section 4.4.3 of MIL-W-5013H. This section details the tests which must be conducted and the

performance requirements which must be obtained to assure that a production brake is suitable for military use. In addition to the MIL-W-5013H tests, the brake will be cycled several times (Seal Break In) to properly seat the brake piston seals.

Each modified KC-135 brake will be subjected to the following individual tests.

1. Examination of Product (per MIL-W-5013H, Section 4.5.1)
2. Seal Break In
3. Proof Pressure and Static Leakage Test (per MIL-W-5013H, Section 4.5.13.1)
4. Dynamic Leakage Test (per MIL-W-5013H, Section 4.5.13.2)

These tests will be conducted using AO-8 hydraulic fluid.

The Examination of Product, Seal Break In and the leakage tests will be performed on each modified brake at ambient laboratory conditions. The dynamic leakage tests will be performed on both modified brakes at ambient temperature and one of the two modified brakes at -65 degrees F and 160 degrees F. Detailed explanations of each test, the test sequence, recorded data, instrumentation etc. are given in following paragraphs.

These tests will be repeated for the unmodified brake assemblies.

3.3 TEST SETUP

The modified brake will be installed in the test setup (shown schematically in Figure 3.1). All testing will be accomplished with this test setup. The number and approximate location of all necessary test instrumentation are shown or described in Figure 3.1.

The test setup will be placed in an environmental chamber for those tests conducted at temperatures other than ambient room temperature.

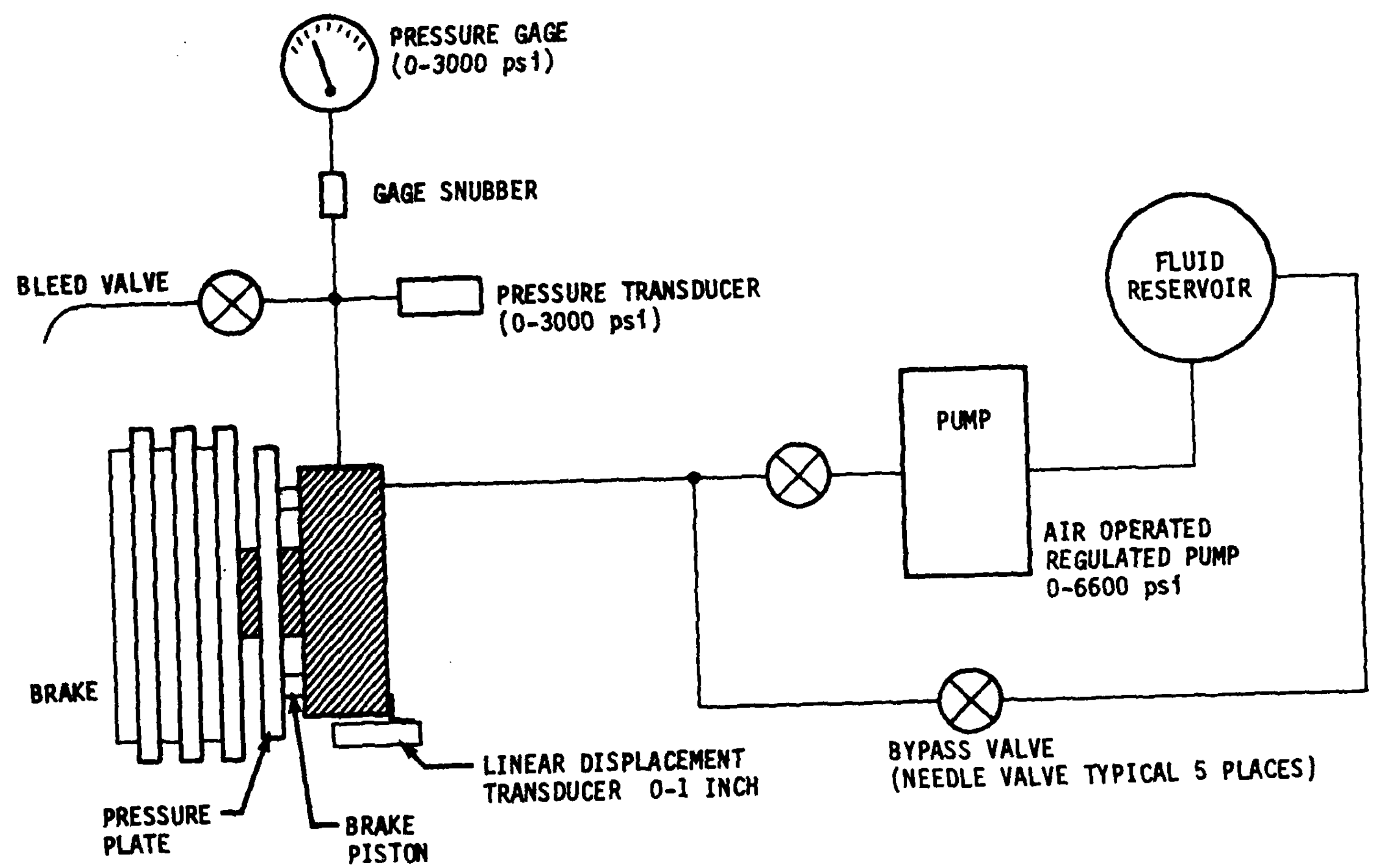

Figure 3.1 Test Setup - Brake, Component Performance Testing

3.4 TEST EQUIPMENT

The equipment and instrumentation used during the tests are listed in Table 3.1. The range, accuracy and resolution required for each piece of equipment used for data acquisition are also listed in the table.

3.5 TEST DESCRIPTIONS

Reference to the test setup schematic (Figure 3.1) will provide additional insight into the test descriptions given below.

3.5.1 EXAMINATION OF PRODUCT, TEST 1

The modification and reassembly of each production KC-135 brake which will be subjected to the AO-8 hydraulic fluid will be supervised and observed by the test engineer to assure that the modified brake conforms to the standards and specifications set by the government for new (or reconditioned) KC-135 brakes. The overhaul and assembly procedures detailed in the Air Force KC-135 Brake Assembly Technical Manual (T.O. 4B1-4-263) will be observed during the modification and reassembly.

3.5.2 SEAL BREAK IN, TEST 2

Prior to functional testing, the brake will be subjected to a break in period of 200 cycles of the application and release of pressure. The objective of this break in period is to assure proper seating of dynamic seals and wear any residual mold release agent or lubricant from the seals prior to the start of testing.

Pressure will be cycled in the sequence defined in the Test Outline (Table 3.2).

TABLE 3.1 TEST EQUIPMENT - BRAKE, COMPONENT PERFORMANCE TESTING

ITEM	RANGE	ACCURACY	RESOLUTION	USE	COMMENT
Pressure Gage	0 to 3000 psi	5 psi	10 psi	Brake Pressure	Indication only
Pressure Teansducer	0 to 3000 psi	5 psi	10 psi	Brake Pressure	
Thermocouple	-100 to 212 F	2°F	1°F	Environment Temperature	
Linear Displacement Transducer	0 to 1 inch	.01 inch	.001 inch	Piston Displacement	
Environmental Chamber Temperature Capability	-100 to 212 F	5°F	—	Temperature Control	
AO-8 Fluid Regulated Pressure Supply Source	0 to 6600 psi	—	—	Fluid Pressure Source	
Brush Chart Recorder 8 Channel	—	—	—	Time History Data	

TABLE 3.2 TEST OUTLINE - BRAKE, COMPONENT PERFORMANCE TESTING

TEST	TEST CONDITIONS		REPETITIONS	RECORDED DATA	BRAKES TESTED*
	PRESS. -psi-	TEMP. -°F-			
1. Examination of Product	---	---	---	Comments as necessary	#1 and #2
2. Seal Break In	0 to 300 0 to 600 0 to 900	Ambient	50 Cycles 50 Cycles 100 Cycles	Comments as necessary	#1 and #2
3. Proof Pressure and Static Leakage Test	0 - 1800	Ambient	1	Brake Pressure Piston Displacement** Fluid Leakage	#1 and #2
4. Dynamic Leadage Test					
4a Ambient	0 - 1200	Ambient	1	Brake Pressure Piston Displacement** Fluid Leakage Release Time **	#1 and #2
4b -65°F	0 - 1200	-65	1		#1
4c 160°F	0 - 1200	160	1		#1

* Two KC-135 brakes will be modified for use in the 2 fluid brake hydraulic system mockup, these brakes will be disignated brake #1 and #2. A similar designation will be used for the unmodified brakes.

** One brake only. Piston displacement and release time will be determined by measuring the displacement of the brake pressure plate.

LEAKAGE TESTS

The static and dynamic leakage tests described below will be performed to assure that the application of the AO-8 hydraulic fluid and the PNF seals in the brake assembly meets the government performance requirements.

3.5.3 PROOF PRESSURE AND STATIC LEAKAGE, TEST 3

Test Objective

Measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under static conditions at proof pressure.

Test procedure

The brake will be pressurized to 1.5 times its maximum operating pressure (1800 psi.) for 5 minutes. The brake pressure shall then be reduced to 5 psi. for 5 minutes.

Performance Requirements

There shall be no measurable leakage (less than one drop) from each brake piston. The brake pistons will return to the retracted position (no permanent set) when pressure is relieved.

3.5.4 DYNAMIC LEAKAGE, TEST 4

Test Objective

Measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under dynamic pressure conditions.

Test Procedure

The brake will be subjected to 25 cycles of the application and release of maximum operating pressure (1200 psi). The brake pistons will be allowed to return to an equilibrium position after each release of pressure and prior to reapplication of pressure position. The time required for the brake piston to reach an equilibrium position will be noted. The brake will be allowed to stabilize at the test temperature before testing (a minimum temperature soak time of 6 hours will be observed).

Performance Requirements

Leakage at static seals shall not exceed a trace. Leakage at dynamic seals shall not exceed one drop of fluid per each 3 inches of peripheral seal length. The pistons shall return to the fully retracted position after each release of pressure.

3.6 TEST CONDITIONS

Each modified KC-135 brake will be tested at ambient laboratory conditions as described in Sections 3.2 thru 3.5. In addition one of the modified brakes will be subjected to the dynamic leakage test (see Section 3.7.2) at temperatures of -65°F and 160°F. The conditions associated with each test are given in the Test Outline, Table 3.2.

3.7 TEST SCHEDULE

The brake tests shall be performed in the sequence listed in the Test Outline, Table 3.2. The data to be recorded during each test and the number of runs per test are given in Table 3.2.

3.8 FLUID SAMPLES

Samples of the AO-8 hydraulic fluid (8-ounce size) will be taken at the beginning and end of the brake component tests and supplied to AFWAL/MLBT.

SECTION IV

SYSTEM PERFORMANCE TESTING

4.1 DISCUSSION OF PROBLEMS AND TEST OBJECTIVE

The conversion of a conventional single-fluid hydraulic brake system to a two fluid system can effect both the dynamic response of the hydraulic system and the stopping performance of the aircraft. Factors such as seal friction, fluid viscosity, fluid density and fluid bulk modulus can change the dynamic response of the hydraulic system thus effecting the stopping performance of the aircraft. The effects which these factors have can be minimized or even eliminated by adjusting hydraulic line sizes and restrictions and by tuning the antiskid system. It is the objective of the system tests to 1) determine the effects which a two-fluid system has upon the dynamic response of the brake hydraulic system, system components and the stopping performance of the aircraft, and 2) determine the brake system modifications necessary to achieve stopping performance comparable to the conventional single-fluid system.

4.2 SYSTEM PERFORMANCE TESTS

The series of brake system performance tests listed in Table 4.1 will be performed to determine and/or evaluate the effects which a two-fluid hydraulic brake system has upon the dynamic response of the brake system and the braking performance of the aircraft. These tests will be performed with mockups of both a KC-135 brake hydraulic system and a two-fluid brake system (modified KC-135 brake hydraulic system). The data generated with the KC-135 brake system will be used to establish baseline or reference dynamic response and braking performance data. To determine and/or evaluate the effects of the two-fluid hydraulic brake system concept, the baseline data will be compared with the two-fluid system data.

The system tests have been divided into two categories, operational characteristics and braking performance tests. The operational characteristics tests are designed to determine the static characteristics and

TABLE 4.1 SYSTEM PERFORMANCE TESTS

OPERATIONAL CHARACTERISTICS TESTS

TEST 1. Frequency Response

TEST 2. Step Response

TEST 3. Static Antiskid Valve Current Versus Brake Pressure

TEST 4. Static Brake Pressure Versus Brake Volume

BRAKING PERFORMANCE TEST

TEST 5. Constant Friction Runway

TEST 6. Wet Runway

TEST 7. Step Friction

TEST 8. Landing Gear System Stability

and dynamic response of the overall hydraulic system and select components within the system. The braking performance tests are designed to determine the stopping performance of the combined KC-135 aircraft and brake system under a variety of environmental (runway friction) conditions. A description of each test including the objective of the test and a detailed test procedure is given in the following paragraphs.

4.3 TEST SETUP

The system tests will be performed using the KC-135 aircraft and brake control simulation. The simulation includes a digital-analog computer model of the aircraft and landing gear systems and a hardware mockup of the brake control system. A schematic of the simulation (test setup) is shown in Figure 4.1. A detailed schematic of the hydraulic brake system mockup is shown in Figure 4.2 to provide additional information concerning the location of test points. System tests 1 thru 4 will be conducted using only the brake hydraulic system mockup (hardware) portion of the simulation. Tests 5 thru 8 will utilize both the computer and hardware portions of the simulation.

4.4 TEST EQUIPMENT

The equipment and instrumentation used during the system testing are listed in Table 4.2. The range, accuracy and resolution required for each piece of equipment used for data acquisition are also listed in the table.

The hydraulic brake system mockup will be placed in an environmental chamber for those tests conducted at temperatures other than ambient room temperature.

4.5 TEST DESCRIPTIONS

Reference to the test setup schematic (Figure 4.2) will provide additional insight into the test descriptions given below.

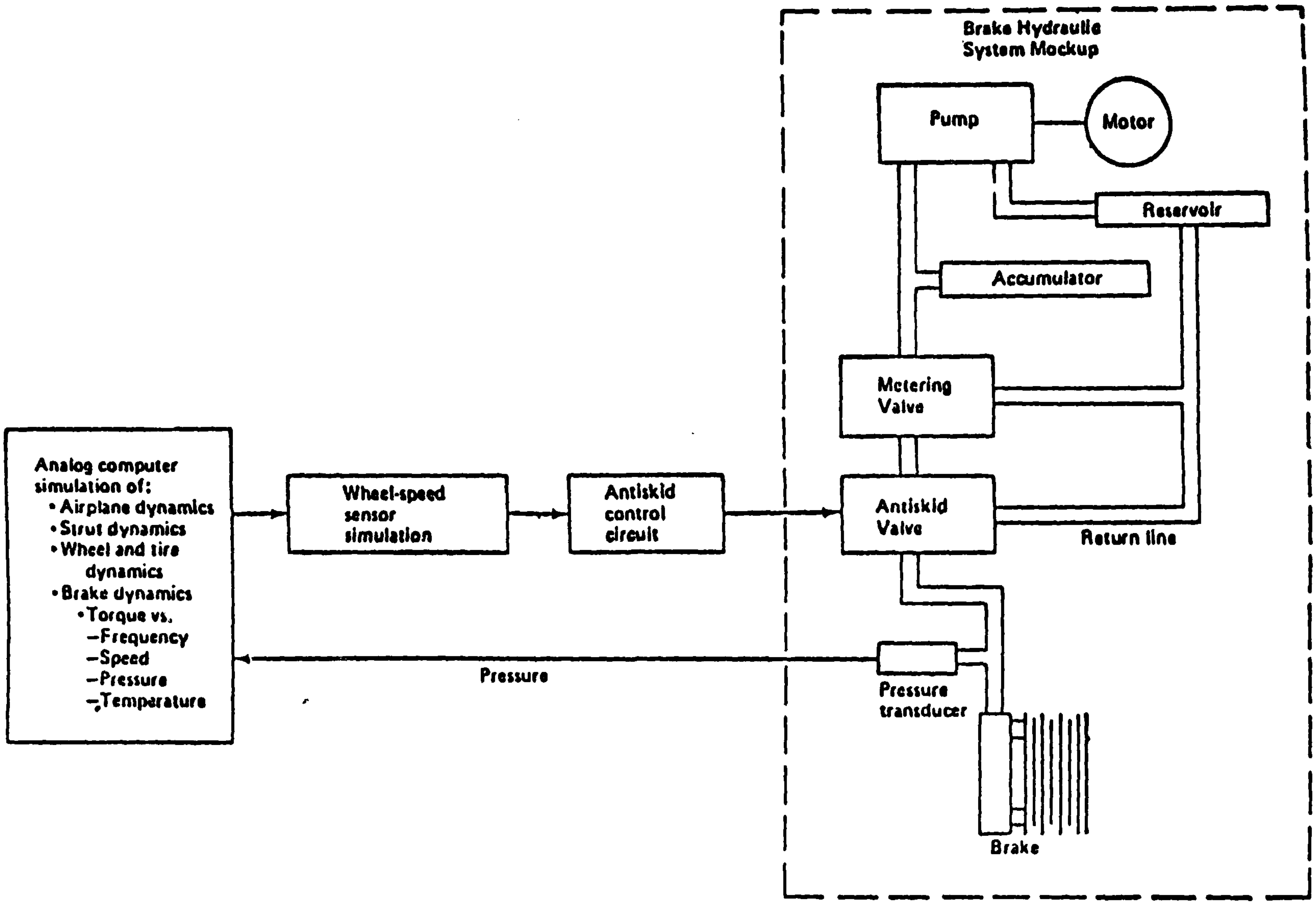

Figure 4.1 Test Setup - System Performance Testing, Hybrid Computer Simulation

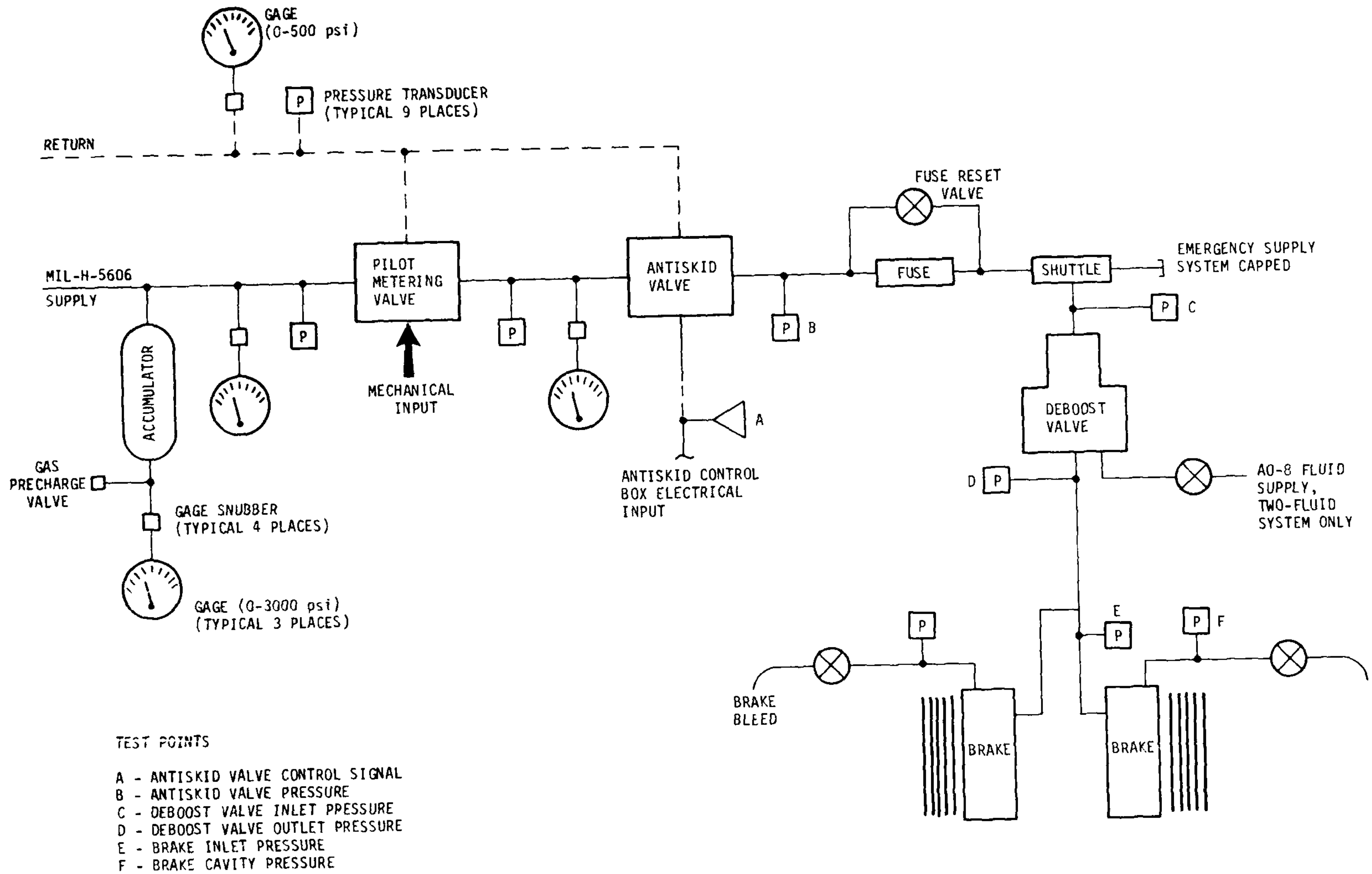

Figure 4.2 Test Setup - Hydraulic Brake System Mockup

AD-A111 319 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 1/2

FIREPROOF BRAKE HYDRAULIC SYSTEM.(U)

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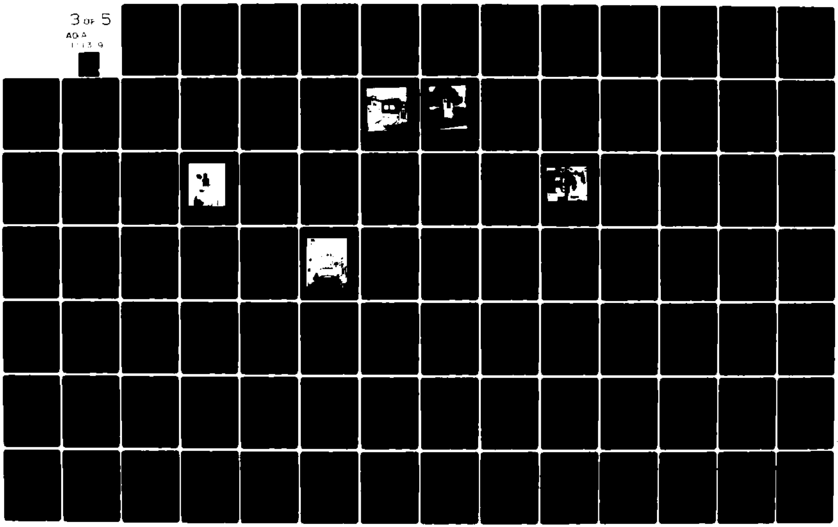


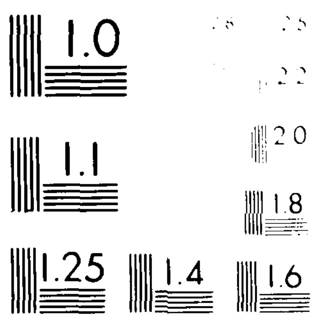

1.0
2.2
2.0
1.1
1.8
1.25
1.4
1.6

TABLE 4.2 TEST EQUIPMENT - SYSTEM PERFORMANCE TESTING

ITEM	QTY.	RANGE	ACCURACY	RESOLUTION	USE	COMMENT
Pressure Gage	3	0 - 3000 psi	5 psi	10 psi	Accumulator, supply and Pilot Metered Pressure	Indication Only
Pressure Gage	1	0 - 500 psi	5 psi	5 psi	Return Pressure	Indication Only
Pressure Transducer	8	0 - 3000 psi	2 psi	1 psi	See Figure 4.2	
Pressure Transducer	1	0 - 500 psi	1 psi	0.5 psi	Return Pressure	
Thermocouple	1	-100 to 212 F	2°F	1°F	Environmental *Temperature*	
Environmental Chamber Temperature Capability	1	-100 to 212 F	5°F	—	Temperature Control	
AO-8 Fluid Supply	1	—	—	—	Fluid	
MIL-H-5606 Fluid Regulated Pressure Supply Source	1	0 - 3000 psi	—	—	Fluid/Pressure Source	

TABLE 4.2 TEST EQUIPMENT - SYSTEM PERFORMANCE TESTING (CONTINUED)

ITEM	QTY.	RANGE	ACCURACY	RESOLUTION	USE	COMMENT
Brush Chart Recorder (8 Channel)	1	—	—	—	Time History Data	
EMR	1	—	—	—	Frequency Response Analysis	
Hybrid Computer Laboratory	1	—	—	—	Aircraft Simulation	Braking Performance Tests

4.5.1 FREQUENCY RESPONSE, TEST 1

Test Objective

Measure the dynamic response (gain and phase angle) of the system and components to a sinusoidal antiskid valve control signal.

Test Procedure

A D.C. electrical control signal corresponding to the desired D.C. pressure level of the test will be applied to the antiskid valve. A 0.5 Hertz sinusoidal electrical control signal will be superimposed on top of the DC signal. The amplitude of the sinusoidal electrical signal will be adjusted until the desired pressure amplitude at the brake is obtained. The frequency of the sinusoidal signal will then be varied between 0.5 Hertz and 50 Hertz. The gain and phase angle of the system and components as defined in Table 4.3 will be determined as a function of frequency. The tests will be performed at laboratory ambient conditions, -65 degrees F and 160 degrees F. The hydraulic system mockup will soak at the test temperature for a minimum of 6 hours prior to testing.

4.5.2 STEP RESPONSE, TEST 2

Test Objective

Measure the dynamic response (time history) of the system or a series of components to a step change in the antiskid valve control signal.

Test Procedure

A D.C. electrical control signal corresponding to the initial test pressure level will be applied to the antiskid valve. The control signal will then be stepped up or down to a level corresponding to the final test pressure level desired after all transients have damped out. Time history plots of the control signal and test pressures as defined in Table 4.3 will be recorded. The tests will be performed at laboratory ambient conditions, -65 degrees F and 160 degrees F. The hydraulic system mockup will soak at the test temperature for a minimum of 6 hours prior to testing.

4.5.3 STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE, TEST 3

Test Objective

Measure the pressure-current characteristic of the antiskid valve. This test is for reference only and will be used to determine whether the antiskid valve meets the manufacturer's specifications.

Test Procedure

A 0.02 Hertz sinusoidal electrical control signal with a current amplitude of 0 to 50 milliamps will be applied to the antiskid valve. Brake pressure will be recorded as a function of the control signal. The test will be performed at ambient laboratory conditions.

4.5.4 STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME, TEST 4

Test Objective

Determine the static brake pressure as a function of the fluid volume contained in the brake. This test is for reference only and will be used to define the characteristics of the brake.

Test Procedure

The brake will be pressured to its maximum operating pressure (965 psi). The pressure supply port of the brake will then be closed. A small quantity of fluid will then be bleed from the brake bleed port into a graduated cylinder. The fluid volume and brake pressure will be recorded. This bleed and recording procedure will be repeated until brake pressure it completely relieved. The test will be performed at ambient laboratory conditions.

4.5.5 CONSTANT FRICTION RUNWAY, TEST 5

Test Objective

Determine the stopping performance of the aircraft in terms of rollout distance under normal runway conditions.

Test Procedure

During this test, braking will be initiated at a typical brake application velocity and continue until the aircraft decelerates to a typical turnoff velocity. The peak available ground friction coefficient (mu) will be held at a constant value throughout the entire run. The distance travelled from brake application to turnoff will be recorded.

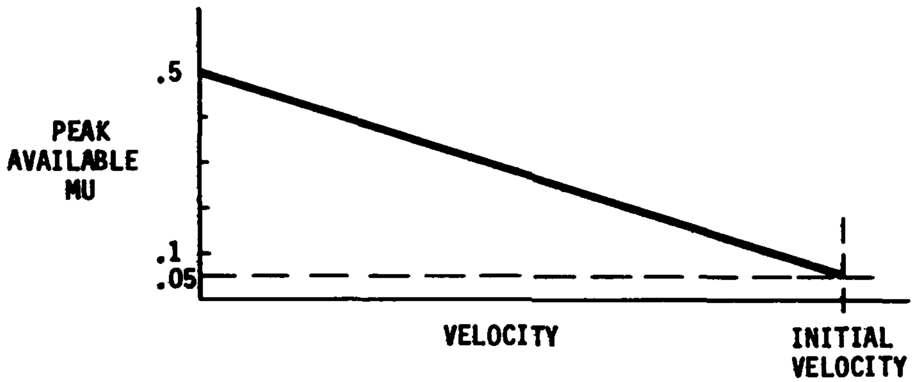
4.5.6 WET RUNWAY, TEST 6

Test Objective

The wet runway test is designed to study the adaptability of the brake control system to slowly changing runway friction conditions.

Test Procedure

During this test, braking will be initiated at a typical brake application velocity and continue until the aircraft decelerates to a typical turnoff velocity. The peak available ground friction coefficient will be made to vary from a low value at high speed to a high value at low speed, see Figure 4.3. This relationship is representative of the available ground mu normally encountered on a wet runway. The distance travelled from brake application to turnoff will be recorded.



WET RUNWAY - TEST 6 FRICTION LEVEL .05 TO .5

Figure 4.3 Wet Runway Test - Friction Versus Speed

4.5.7 STEP FRICTION, TEST 7

Test Objective

The step friction test is designed to study the adaptability of the brake control system to rapidly changing runway friction conditions.

Test Procedure

During this test, braking will be initiated at a typical brake application velocity and continue until the aircraft decelerates to a typical turnoff velocity. The peak available ground friction coefficient will be made to vary in step fashion as shown in Figure 4.4. Several step changes will be made during the braking run, so that system operation can be observed under a variety of conditions. The distance travelled from brake application to turnoff will be recorded.

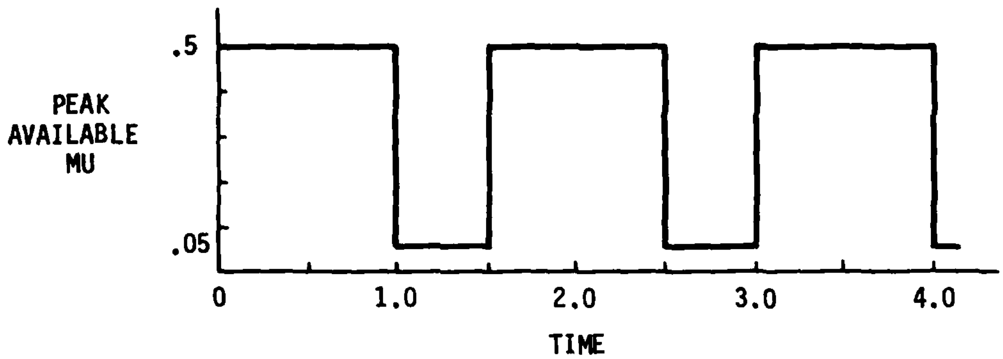
4.5.8 LANDING GEAR SYSTEM STABILITY, TEST 8

Test Objective

The stability test is designed to measure the ability of the brake control system to contribute to the fore and aft vibrational stability of the landing gear.

Test Procedure

The stability margin of the system will be determined by establishing the amount of strut damping required for stable landing gear oscillations. During a normal braking run the landing gear strut will be made to oscillate by increasing the brake torque to 1.5 times is normal value for a short period of time (i.e. a brake torque impulse). The brake torque impulses will be applied at various velocities so the strut oscillations can be observed at a variety of conditions. The strut damping will be lowered until the landing gear oscillations are no longer damped, the brake system goes unstable or strut damping is zero. The strut displacement as a function of time will be recorded in addition to the strut damping ratio.



0.5 SECOND FRICTION STEP IS REPEATED EVERY SECOND AS SHOWN

Figure 4.4 Step Friction Test - Friction Versus Time

4.6 TEST CONDITIONS

The test conditions associated with each system test are given in the System Performance Test Outline, Table 4.3. The values of parameters varied during each test are specified in the test outline.

Tests are to be performed on a unmodified system with MIL-H-5606 hydraulic fluid and with a modified system with the two fluids.

4.7 TEST SCHEDULE

The system tests will be performed in the sequence listed in the System Performance Test Outline, Table 4.3. The data to be recorded during each test and the number of runs per test are also given.

4.8 FLUID SAMPLES

Samples of the AO-8 hydraulic fluid (8-ounce size) will be taken periodically and provided to AFWAL/MLBT throughout the system tests. The samples will be taken after 0, 2 and 5 hours of testing and at 5 hour increments thereafter (not including temperature soak time).

TABLE 4.3 TEST OUTLINE - SYSTEM PERFORMANCE TESTING

TEST 1 FREQUENCY RESPONSE

TEST	TEST TEMPERATURE	TEST POINTS (FIGURE 4.2)		FREQUENCY RANGE (HZ)	TEST CONDITION PRESSURE MEASURED AT BRAKE	TEST REPETITIONS FOR AVERAGING AT EACH TEMPERATURE
		INPUT	OUTPUT			
1a. Brake System	Ambient -65°F 160°F	A	E	0.5 - 50	33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi	3 3 3 3
1b. Antiskid Valve	Ambient -65°F 160°F	A	B		33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi	3 3 3 3
1c. Deboost Valve	Ambient -65°F 160°F	C	D		33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi	3 3 3 3
1d. Brake	Ambient -65°F 160°F	E	F		33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi	3 3 3 3

TABLE 4.3 TEST OUTLINE - SYSTEM PERFORMANCE TESTING (CONTINUED)

TEST 2 STEP RESPONSE

TEST	TEST TEMPERATURE	TEST POINTS INPUT	TEST POINTS OUTPUT	TEST CONDITION-% OF FULL BRAKE PRESSURE	TEST REPETITIONS FOR AVERAGING AT EACH TEMPERATURE
2a. Increasing Pressure Step	Ambient	A	B,C,D,F	0 - 50	3
	-65°F			0 - 80	3
	160°F			0 - 100	3
				20 - 50	3
				20 - 80	3
				20 - 100	3
				50 - 80	3
				50 - 100	3
2b. Decreasing Pressure Step	Ambient	A	B,C,D,F	50 - 0	3
	-65°F			80 - 0	3
	160°F			100 - 0	3
				50 - 20	3
				80 - 20	3
				100 - 20	3
				80 - 50	3
				100 - 50	3

TEST 3 STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE

	Ambient	A	F	33	3
				66	3
				100	3

TABLE 4.3 TEST OUTLINE - SYSTEM PERFORMANCE TESTING (CONTINUED)

TEST 4 STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME

TEST TEMPERATURE	RECORDED DATA	TEST REPETITIONS FOR AVERAGING
Ambient	BRAKE FLUID VOLUME AND BRAKE PRESSURE	3

TEST 5 CONSTANT RUNWAY FRICTION

TEST TEMPERATURE	FRICTION LEVEL*	RECORDED DATA	TEST REPETITIONS FOR AVERAGING
Ambient	.6	Stopping Diatance	3
-65°F	.5	and Other Key	3
160°F	.4	Simulation Parameters	3
	.3		3
	.2		3
	.1		3
	.075		3
	.05		3

TEST 6 WET RUNWAY - FRICTION AS A FUNCTION OF AIRCRAFT VELOCITY (See Figure 4.3)

TEST TEMPERATURE	FRICTION LEVEL*	RECORDED DATA	TEST REPETITIONS FOR AVERAGING
Ambient	.05 - .5	Stopping Distance	3
-65°F	.05 - .35	and Other Key	3
160°F		Simulation Parameters	

TABLE 4.3 TEST OUTLINE - SYSTEM PERFORMANCE TESTING (CONTINUED)

TEST 7 STEP FRICTION - (See Figure 4.4)

TEST TEMPERATURE	FRICTION LEVEL*	RECORDED DATA	TEST REPETITIONS FOR AVERAGING
Ambient -65°F 160°F	.05 - .5	Stopping Distance and Other Key Simulation Parameters	3

* FRICTION LEVELS MAY BE REVISED SLIGHTLY BASED ON OPERATIONAL CAPABILITY OF BASELINE SYSTEM.

TABLE 4.3 TEST OUTLINE - SYSTEM PERFORMANCE TESTING (CONTINUED)

TEST 8 LANDING GEAR SYSTEM STABILITY

The ability of the brake control system to contribute to the stability of the landing gear will be evaluated. The stability margin of the system will be determined by establishing the amount of strut damping required for stability.

The nominal damping ratio for the KC-135 strut is 0.1. Damping will be reduced in increments of .01 until the strut or system goes unstable or strut damping is zero.

TEST TEMPERATURE	DAMPING RATIO	RECORDED DATA	TEST REPETITIONS FOR AVERAGING AT EACH TEMPERATURE
Ambient	.1	Fore and Aft	3
-65°F	.09	Strut Displacement	3
160°F	.08		3
	.07		3
	.06		3
	.05		3
	.04		3
	.03		3
	.02		3
	.01		3
	0		3

APPENDIX C-1

PRODUCTION DEBOOST VALVE TEST PROCEDURE

The manufacturer's recommended test procedure and test setup for acceptance testing of production deboost valves are given in Table C.1 and Figure C.1.

TABLE C.1 PRODUCTION DEBOOST VALVE ACCEPTANCE TEST PROCEDURE

FUNCTIONAL TEST REQUIREMENTS

GENERAL

Functional test shall be conducted at room temperature with MIL-O-5606 hydraulic fluid. After testing flush with MIL-O-6083 preservative oil. DO NOT DRAIN. Cap all ports and tag with test date. As an option MIL-O-6083 oil may be used to conduct the functional test.

DO NOT use compressed air on the ports at any time.

Remove screen assembly 9-65813 so piston O-ring leakage and piston position may be observed.

TEST

1. Apply pressure at port "B" to "bottom" piston at small end. Piston can be observed through port "A." Bleed the deboost valve by continuing flow through port "A."

2. With the piston "bottomed" at the small end, plug port "A." Apply 1445 PSI proof pressure at port "B" and hold for two minutes. Reduce pressure to 5 PSI and hold for two minutes. There shall be no evidence of external leakage or permanent set.

3. CAUTION: Use static pressure only for this test. With port "B" open apply pressure at port "A." The piston shall move to the large end and "bottom" as indicated by continuous flow from port "B." With the piston maintained in this position, install 1500 PSI relief valve and gage at port "B" and apply 4500 PSI to port "A" and hold for two minutes. Reduce pressure to 5 PSI and hold for two minutes. There shall be no evidence of external leakage or permanent set.
   WARNING: If excessive leakage past the ball check is allowed to move the piston toward the small end and "bottom," application of 4500 PSI will rupture the large end of the deboost valve. Remove pressure at port "A" immediately if piston becomes visible through vent holes.

INTERNAL LEAKAGE

4. Install the deboost valve in a test set up as shown in Figure C.1. Position the deboost valve so that port "A" is up.

5. With the needle valve open and the 3-way valve poisitioned to permit flow from port "A" to return apply hydraulic pressure at the needle valve. Increase flow until the pressure gage at port "B" reads 1015 (+50/-0) PSI (flow required will be 2.5 to 5 GPM).

6. While flow required in item 5 is maintained, close the needle valve and immediately move 3-way valve to apply 3000 PSI at port "A."
   The pressure gauge should indicate 963 (+50/-50) PSI. Hold 3000 PSI at port "A." Leakage through the ball check will be indicated by slowly rising pressure at port "B" followed by the relief valve cracking (1200 ± 25 PSI). Leakage at the relief valve shall not exceed 15 drops in 10 minutes.

7. With port "B" open and pressure applied to port "A," the piston shall move smoothly toward the large end. Reverse the connection, apply pressure to port "B" when port "A" is open and the piston shall move smoothly toward the small end.

8. Piston seal leakage as observed through the breather holes, shall not exceed one drop per 25 cycles.

9. Replace screen assembly 9-65813 and retorque 6-83807 studs to 25-35 in-lbs maximum.

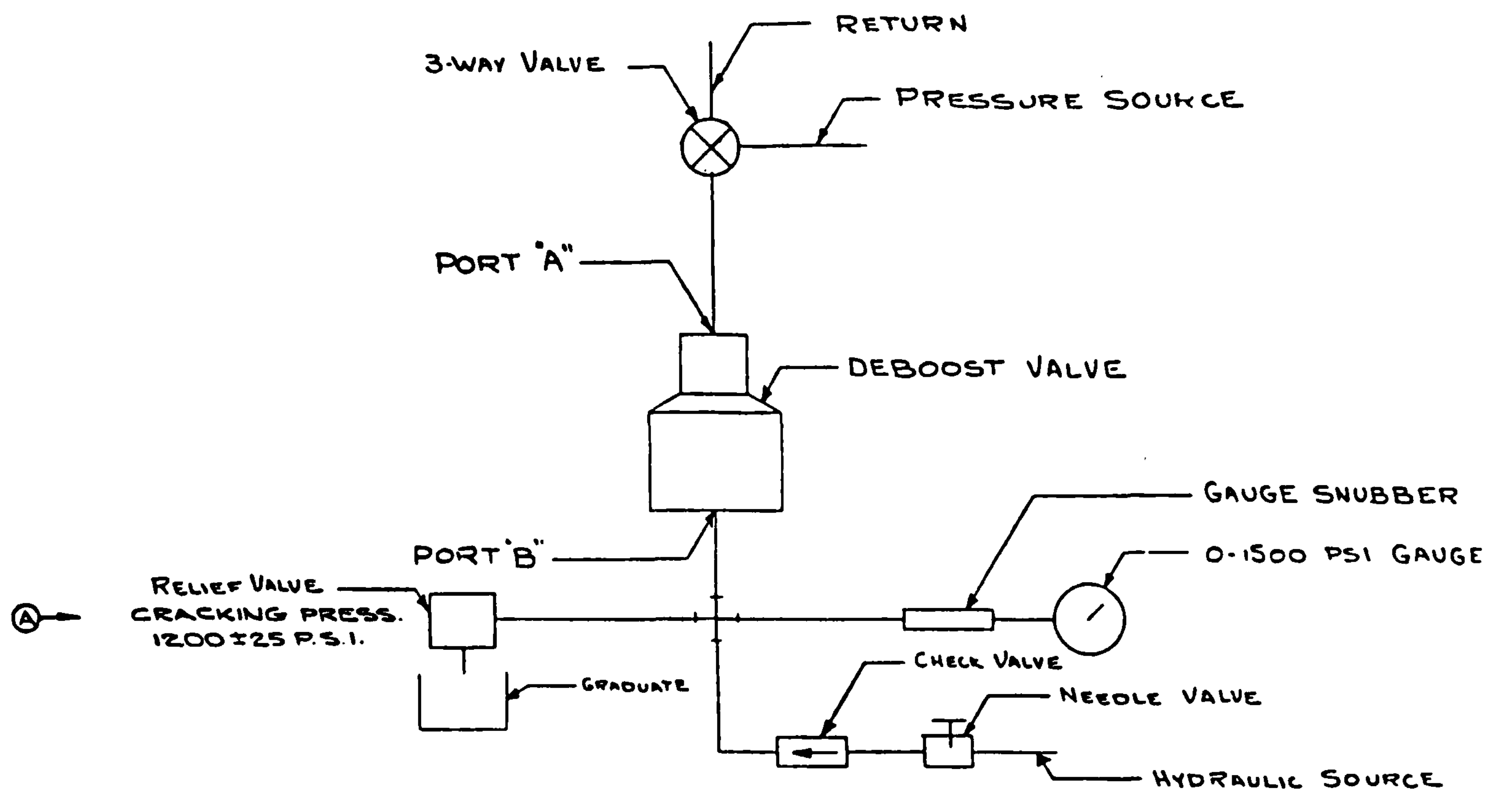


Figure C.1 Test Setup - Production Deboost Valve Acceptance Testing

APPENDIX D

COMPONENT PERFORMANCE TEST RESULTS

D.1 COMPONENT TESTING

Component performance tests were conducted on the modified KC-135 deboost valve, two modified KC-135 brake assemblies, a standard KC-135 deboost valve and two standard KC-135 brake assemblies to assure that each component met the production part performance requirements prior to installation and use of the parts in the CTFE (modified components) and the MIL-H-5606 (baseline components) hydraulic brake system mockups. Each component was subjected to a series of tests which included examination of product, seal break-in, proof pressure, leakage and friction tests. A complete description of each test, the test procedure, requirements, etc., is given in the Component Performance Test Plan, Appendix C. The results of the component tests are summarized in following paragraphs.

The component performance tests were conducted a three temperatures, ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit. Each test set up was placed inside the environmental chamber shown in Figure D-1. The test setup was then soaked at temperature for a minimum of 6 hours prior to testing. The chamber is a microprocessor controlled unit which can be programmed for automatic operation. Two holes, which are plugged with foam insulation during operation, in the bottom of the chamber provide access for instrumentation and fluid power.

D.2 STANDARD KC-135 DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

Component performance testing of the standard KC-135 deboost valve was conducted with a production unit, serial number 2524W. No modifications to the component were required for the tests. The deboost valve was examined and installed in test setup (see Figures D-2 and D-3). The setup was placed in the environmental chamber, filled with MIL-H-5606 hydraulic fluid and bled in preparation for testing.

Figure D-1 Environmental Chamber and Instrumentation

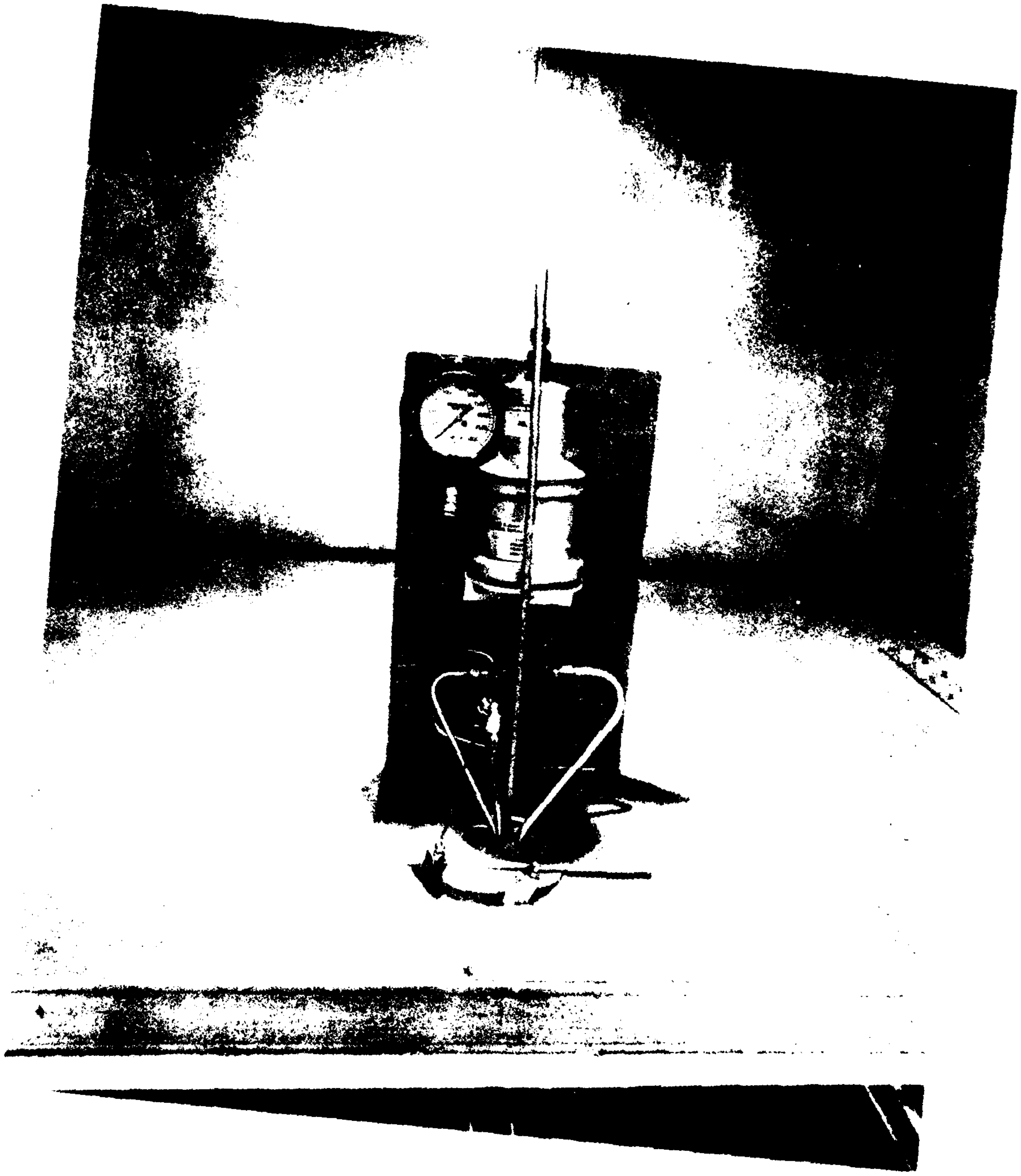


Figure D-2 Standard KC-135 Deboost Valve Test Setup

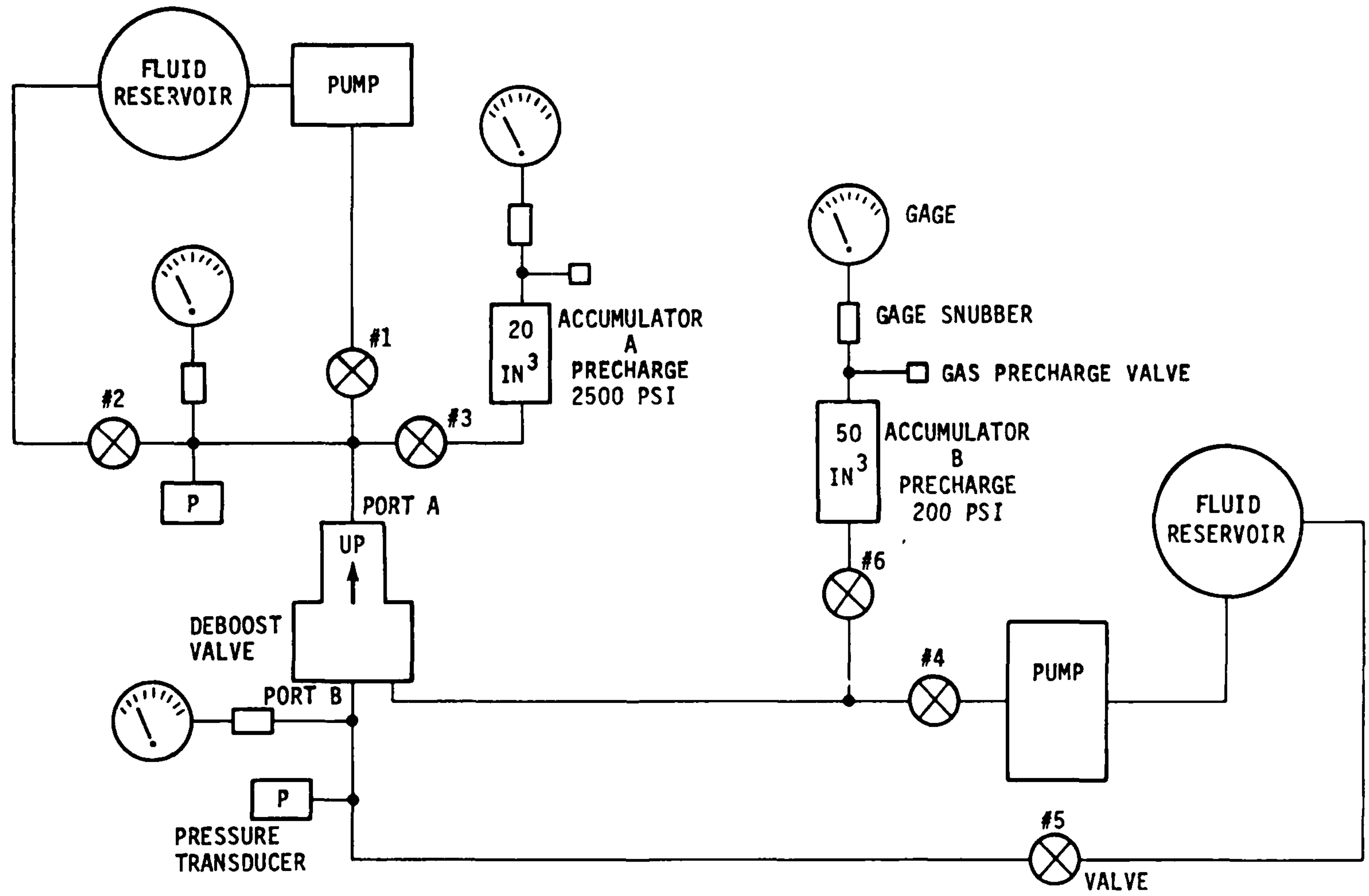


Figure D-3 Deboost Valve Test Setup Schematic

The standard deboost valve met all the test requirements established in the Component Performance Test Plan. The results of the component tests are summarized in Table D-1. A brief description of each test and the results is given in the following paragraphs.

The ambient temperature tests were performed on January 26 and 27, 1981. The temperature in the test area was 67 degrees Fahrenheit.

The low temperature tests were performed on January 29, 1981. The deboost valve was soaked in the environmental chamber for approximately 6 hours and 10 minutes at -65 degrees Fahrenheit prior to starting the tests.

The high temperature tests were performed on January 28, 1981. The deboost valve was soaked in the environmental chamber for approximately 7 hours at 160 degrees Fahrenheit prior to starting the tests.

D.2.1 EXAMINATION OF PRODUCT, TEST 1

The deboost valve was received fully assembled. The valve was not disassembled for examination; however a visual inspection of the unit was made. The valve appeared to be in good condition. No damage or points of leakage were found. The screen covering the vent holes was removed so the deboost piston could be observed during this and subsequent tests.

D.2.2 SEAL BREAK IN, TEST 2

The seal break in test was performed to assure proper seating of dynamic seals prior to the start of testing. The dynamic seals were seated by cycling the deboost piston up and down inside the valve.

The deboost valve was subjected to a break in period of 200 cycles of the application and release of pressure as described in the test procedure. The test was performed at ambient temperature only.

The deboost valve was examined during and at the end of the testing. No fluid leakage was observed. The piston was viewed through the vent hole several times during the test to verify that the piston was being cycled up and down.

TABLE D-1 STANDARD KC-135 DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEMP.	RESULTS AND COMMENTS
1. Examination of Product	---	--	Assembly Inspected Visually, Not Disassembled To Inspect Internal Parts
2. Seal Break In	50 Cycles,0-300 psi 50 Cycles,0-600 psi 100 Cycles,0-965 psi	Ambient	No Leakage
3. Proof Pressure and High Pressure Seal Static Leakage	4500 psi at Port A for 2 Minutes	Ambient	No Leakage
4. Proof Pressure and High Pressure Seal Static Leakage	1445 psi at Port B for 2 Minutes	Ambient	No Leakage
5. Dynamic Leakage			
5a Ambient	25 Cycles,0-965 psi	Ambient	No Leakage
5b -65°F		-65°F	No Leakage
5c 160°F		160°F	No Leakage
6. Seal Friction			
6a Ambient	1500 psi at Port A	Ambient	No Leakage Seal Friction -102 lb
6b -65°F		-65°F	No Leakage Seal Friction 4 lb
6c 160°F		160°F	No Leakage Seal Friction -27 lb

D.2.3 PROOF PRESSURE AND HIGH PRESSURE SEAL STATIC LEAKAGE, TEST 3

The proof pressure and high pressure seal static leakage test was performed to measure the fluid leakage past the high pressure piston seal under static conditions at proof pressure.

Port A of the deboost valve was pressurized to 4500 psi (the pressure at Port B was approximately 1450 psi). Pressure was held for 2 minutes and then reduced to atmospheric. The test was performed at ambient temperature only.

No signs of fluid leakage were observed during or after the test. However, high pressure (4500 psi) could not be maintained during the test. The pressure at Port A decreased approximately 150 psi during the 2 minute duration. Similarily the pressure at Port B decreased about 50 psi (indicative of the 3.11 deboost valve piston area ratio). Since no external leakage was observed and the pressure in both volumes decreased it is suspected that there was some leakage through one of the valves in the test setup.

D.2.4 PROOF PRESSURE AND LOW PRESSURE SEAL STATIC LEAKAGE, TEST 4

The proof pressure and low pressure seal static leakage test was performed to measure the fluid leakage past the low pressure piston seal under static conditions at proof pressure.

Port B was pressurized to 1445 psi (the pressure at Port A was atmospheric and the deboost valve piston bottomed at the small end). The pressure was held for 2 minutes and then reduced to atmospheric. The test was performed at ambient temperature only.

No signs of fluid leakage were observed during or after the test. No droop or sag in pressure over the 2 minutes was observed as noted in Test 3.

D.2.5 DYNAMIC LEAKAGE, TEST 5

The dynamic leakage test was performed to measure the fluid leakage past the piston seals under dynamic pressure conditions.

The deboost valve was subjected to 25 cycles of the application and release of pressure. The test procedure described in paragraph 2.5.4 of Appendix C was changed to prevent fluid flow from low pressure to high pressure through the replenishment valve. The pressure at Port A was cycled between 0 and 3000 psi. The pressure at Port B changed from 0 to 965 (indicative of the deboost valve area ratio) during each cycle. The test was performed at ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit.

No fluid leakage was observed during or after the tests.

D.2.6 SEAL FRICTION, TEST 6

The seal friction test was performed to measure the seal friction load which must be overcome before piston motion occurs.

Seal friction was determined by measuring the deboost valve pressures at Port A and Port B before and after piston motion. Using the data and the piston force balance equations below the seal friction forces F1, F2 and F were determined.

$$F1 = PA3 \times 3.5332 - PB3 \times 11.0270$$
$$F2 = PA5 \times 3.5332 - PB5 \times 11.0270$$
$$F = (PB5 - PB3) \times 11.0270 - (PA5 - PA3) \times 3.5332 = F1 - F2$$

The test was performed at ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit.

The results of the seal friction test are given in Table D-2. The average friction force (F) was measured to be approximately -102 pounds at ambient temperature, +4 pounds a -65 degrees Fahrenheit and -27 pounds at 160 degrees Fahrenheit. Figure D-4 is included to help interpert the data presented in the table. The idealized hydraulic force acting on each side of the piston (PA X 3.5332 and PB X 11.0270) at each step of the test procedure are plotted along with the friction force.

Examination of the test results indicates that there was a significant variation in the measured value of the friction force. Also, comparison of the data with the idealized results reveals that there are sign differences

TABLE D-2 SEAL FRICTION, STANDARD KC-135 DEBOOST VALVE

TEMPERATURE DEGREES F	RUN	PRESSURE, PSI PA3	PA5	PB3	PB5	FORCE, lbs F1	F2	F
AMBIENT, 70°F	1	1524	1594	496	505	- 85	63	-148
	2	1547	1587	503	509	- 81	-6	- 75
	3	1535	1573	500	504	- 90	0	- 90
	4	1538	1575	497	499	- 46	62	-108
	5	1575	1604	507	509	- 26	54	- 80
	6	1592	1629	512	514	- 21	88	-109
						AVERAGE		-102
-65°F	1	1491	1534	523	534	-499	-468	- 31
	2	1534	1561	518	522	-292	-241	- 51
	3	1512	1525	503	509	-204	-224	20
	4	1543	1556	512	519	-194	-225	31
	5	1533	1539	509	512	-196	-208	14
	6	1480	1484	494	497	-218	-237	19
	7	1539	1544	510	514	-186	-212	26
						AVERAGE		4
160°F	1	1562	1619	505	515	- 50	41	- 91
	2	1574	1601	508	511	- 40	22	- 62
	3	1536	1544	498	501	- 64	- 69	5
	4	1500	1509	487	490	- 70	- 72	2
	5	1470	1500	477	484	- 66	- 37	- 29
	6	1499	1513	486	491	- 63	- 69	6
	7	1502	1527	488	494	- 74	- 52	- 22
						AVERAGE		- 27

F1 = PA3 X 3.5332 - PB3 X 11.0270
F2 = PA5 X 3.5332 - PB5 X 11.0270
F = (PB5 - PB3) X 11.0270 - (PA5 - PA3) X 3.5332 = F1 - F2

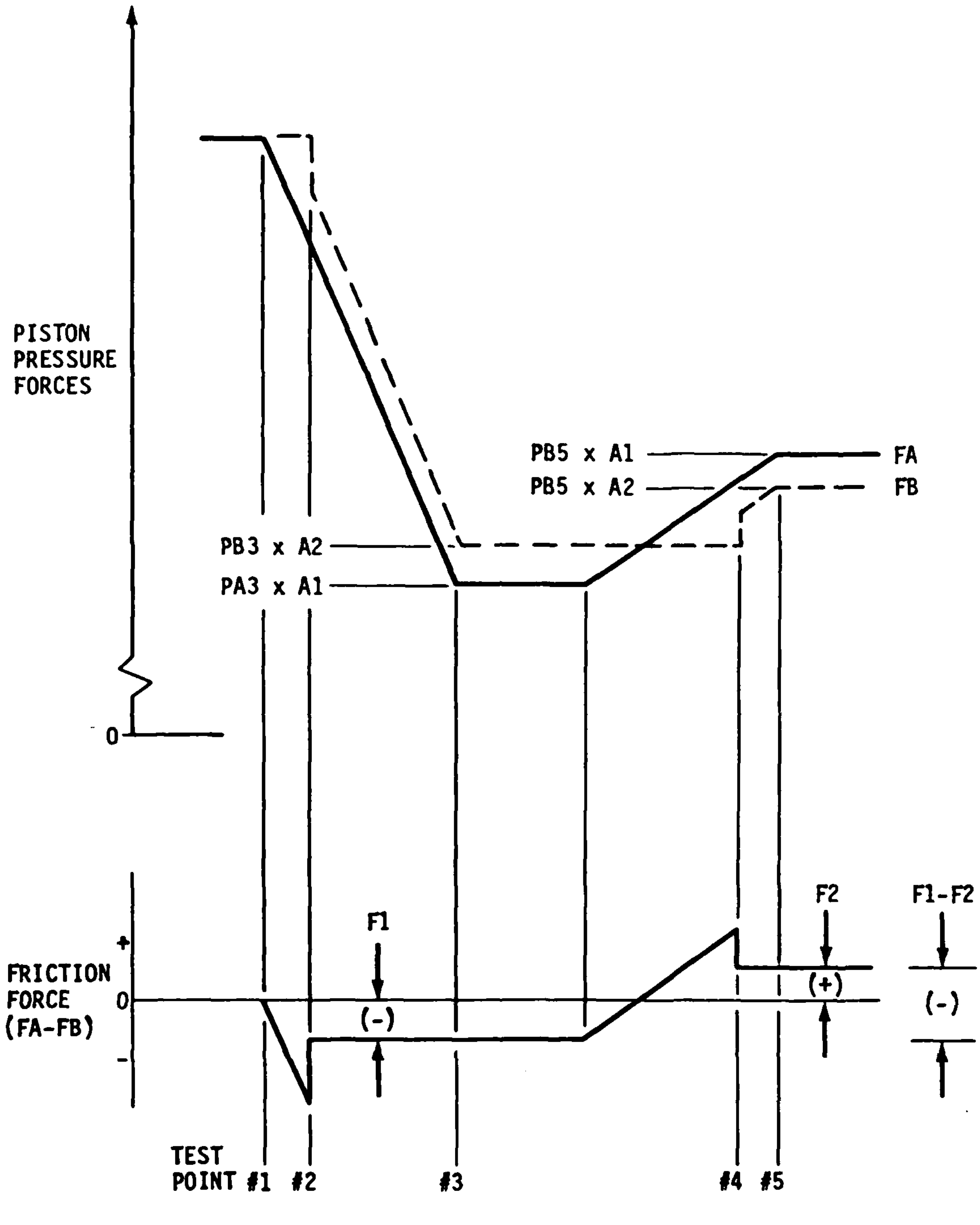


Figure D-4 Idealized Seal Friction

(i.e. the friction acts in the direction opposite to that which was expected). The sign of F1 should be negative, F2 positive and F negative. The test procedure, test setup and instrumentation accuracy were studied to determine the reasons for the data scatter and sign errors. No problems associated with the test procedure or setup were found. However, the hysteresis error in the pressure transducer measurement was found to be a possible source of error. The hysteresis in the pressure transducers (approximately 2.7 psi) results in errors of -20 lbs in F1 and F2 and -40 lbs in F (the measured value is 20 or 40 pounds lower than the actual). Comparing the magnitudes of the hysteresis error to the measured value of friction it can be seen that they are of the same magnitude making it difficult to accurately measure the friction.

To determine the significance of the deboost valve piston seal friction a frequency response computer analysis using HSFR was performed. The results of the analysis are shown in Figure D-5. Three levels of piston seal friction were analyzed; 0,100 and 200 pounds. The seal friction force (F1 or F2) present in the valve was estimated using industry accepted practices, (see Reference 4) to be 110 pounds. It can be seen in Figure D-5 that deboost valve piston friction has little effect on the frequency response of the brake system. Doubling the normal friction (200 pound) has virtually no affect on the system response. Efforts to improve measurement of seal friction were not undertaken since seal friction does not appear to be significant.

D.3 MODIFIED DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

A KC-135 deboost valve modified for use as a fluid isolator was tested during the deboost valve component performance tests. The deboost valve was modified as described in Appendices A and B. The modified and fabricated parts were examined prior to and during assembly. The unit was then installed in the test setup, Figures D-6 and D-3. The setup was placed in the environmental chamber, filled with MIL-H-5606 and CTFE fluids and bled in preparation for testing.

The modified deboost valve met all the performance requirements established in the Component Performance Test Plan. The results of the component tests are summarized in Table D-3. A brief description of each test and the results is given in the following paragraphs.

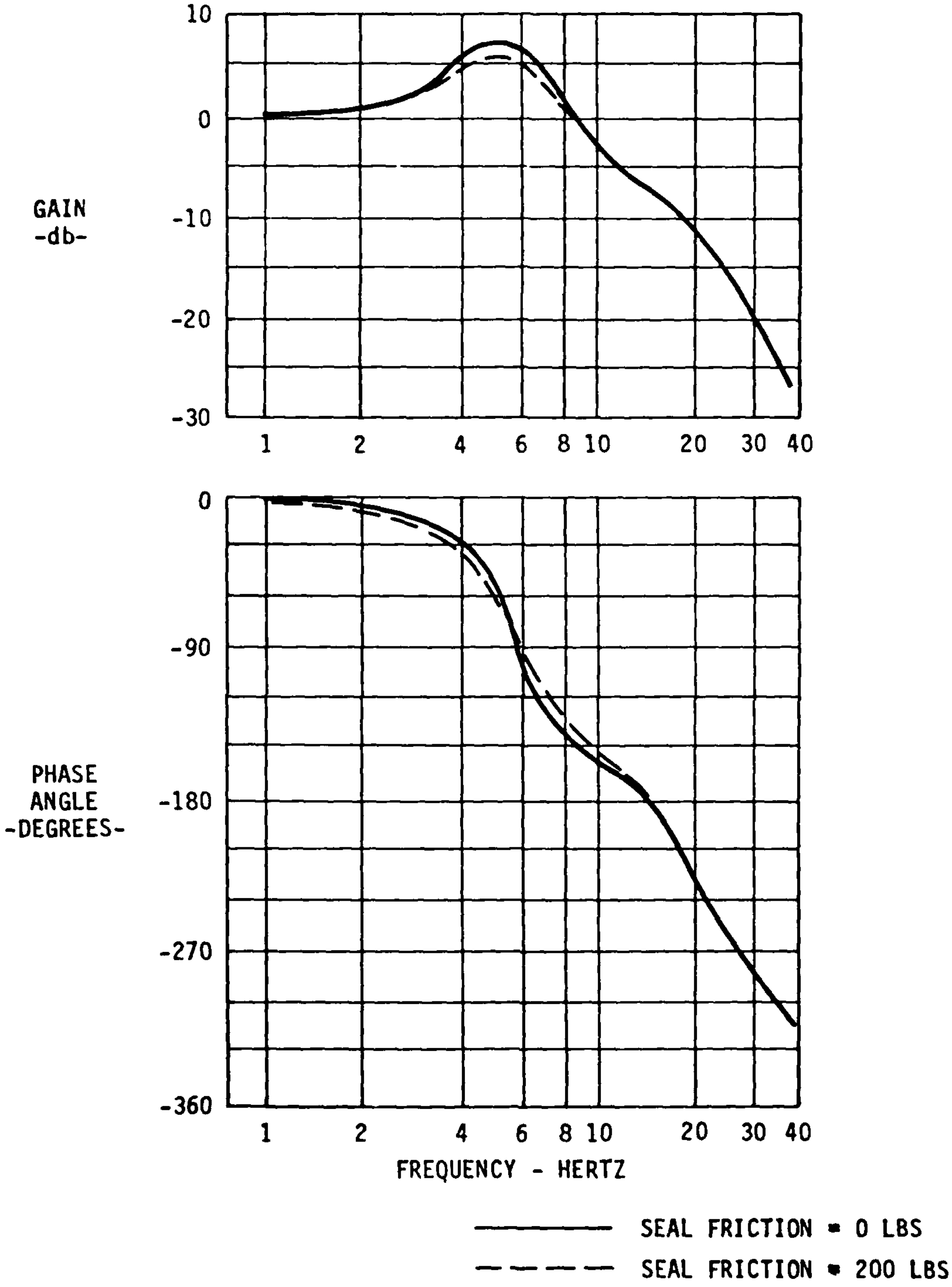


Figure D-5 HSFR Analysis of Seal Friction

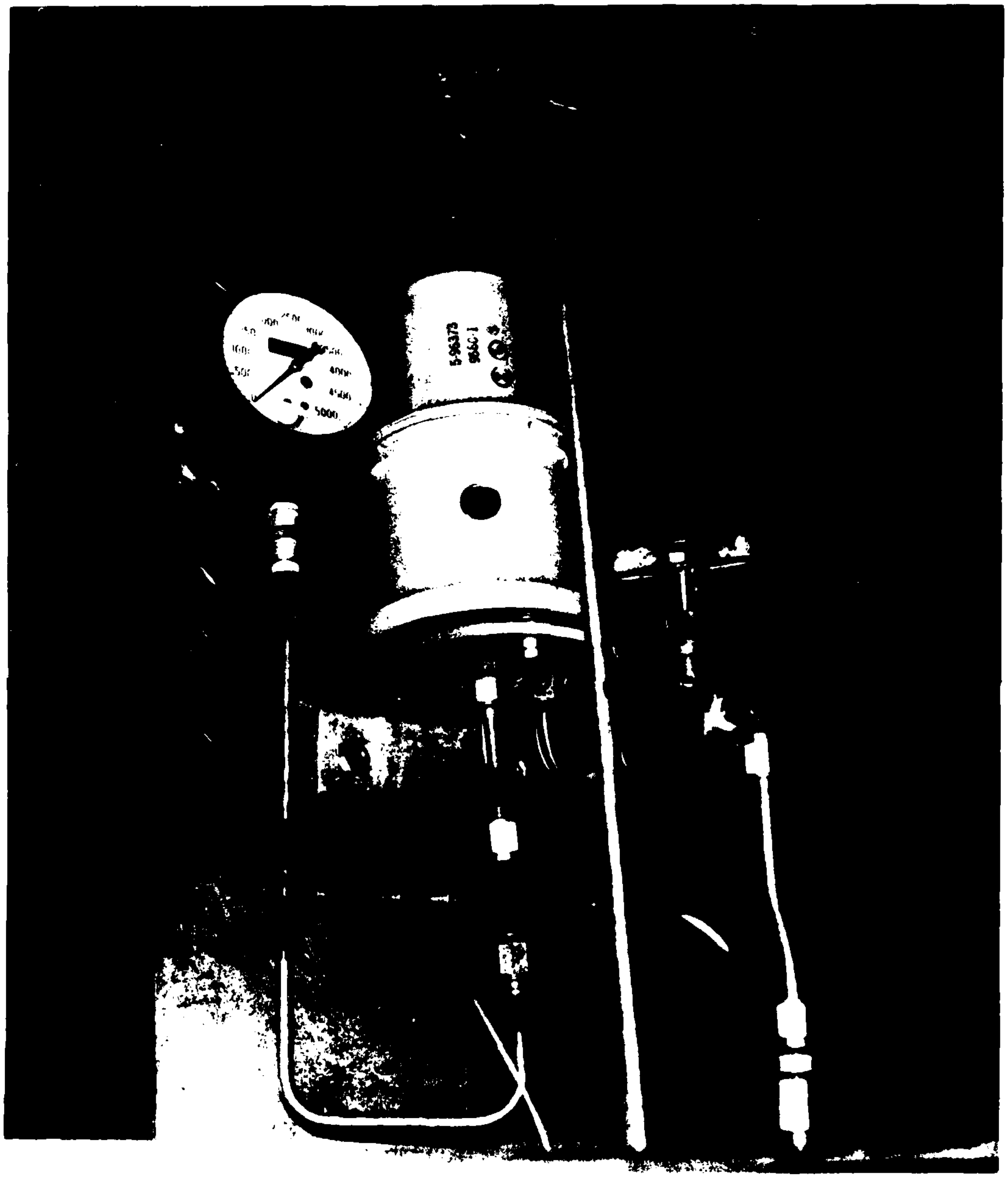


Figure D-6 Modified Deboost Valve Test Setup

TABLE D-3 MODIFIED DEBOOST VALVE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEMP.	RESULTS AND COMMENTS TWO-FLUID
1. Examination of Product	—	—	Manufactured and Modified Parts Visually Inspected, Assembly Observed
2. Seal Break In	50 Cycles,0-300 psi 50 Cycles,0-600 psi 100 Cycles,0-965 psi	Ambient	No Leakage
3. Proof Pressure and High Pressure Seal Static Leakage	4500 psi at Port A for 2 Minutes	Ambient	No Leakage
4. Proof Pressure and High Pressure Seal Static Leakage	1445 psi at Port B for 2 Minutes	Ambient	No Leakage
5. Dynamic Leakage			
5a Ambient	25 Cycles,0-965 psi	Ambient	No Leakage
5b -65°F		-65°F	No Leakage
5c 160°F		160°F	No Leakage
6. Seal Friction			
6a Ambient	1500 psi at Port A	Ambient	No Leakage Seal Friction -90 lb
6b -65°F		-65°F	No Leakage Seal Friction -2 lb
6c 160°F		160°F	No Leakage Seal Friction -69 lb

The ambient temperature tests were performed on March 16, 1981. The temperature in the test area was 65 degrees Fahrenheit.

The low temperature tests were performed on March 20, 1981. The modified deboost valve test setup was soaked in the environmental chamber for approximately 6 hours and 18 minutes at -65 degrees Fahrenheit prior to the start of testing.

The high temperature tests were performed on March 23, 1981. The modified deboost valve test setup was soaked at 160 degrees Fahrenheit for approximately 6 hours and 15 minutes prior to starting the tests.

D.3.1 EXAMINATION OF PRODUCT, TEST 1

The modified and fabricated deboost valve parts were inspected visually prior to and during assembly of the modified deboost valve. All parts and seals were new and appeared to be in good condition. The assembled unit was installed in the test setup and then serviced with fluid. No points of leakage were found.

D.3.2 SEAL BREAK IN, TEST 2

The seal break in test was performed to assure proper seating of the PNF and nitrile seals prior to the start of testing. The seals were seated by cycling the deboost valve piston up and down inside the valve 200 times. The procedure defined in the test plan (Appendix C) was followed during the test. The test was performed at ambient temperature only.

The deboost valve was examined during and at the end of testing. No fluid leakage was observed.

D.3.3 PROOF PRESSURE AND HIGH PRESSURE SEAL STATIC LEAKAGE, TEST 3

The proof pressure and high pressure seal static leakage test was performed to measure the fluid leakage past the high pressure piston seal under static conditions at proof pressure.

Port A of the deboost valve was pressurized to 4500 psi (the deboost valve piston was bottomed at the large end, 0.0 psi at Port B). Pressure was held for 2 minutes and then reduced to atmospheric. The test was performed at ambient temperature only.

No fluid leakage was noted during or after the test. In addition, no significant pressure sag as noted in the standard deboost valve test was observed.

D.3.4 PROOF PRESSURE AND LOW PRESSURE SEAL STATIC LEAKAGE, TEST 4

The proof pressure and low pressure seal static leakage test was performed to measure the fluid leakage past the low pressure piston seal under static conditions at proof pressure.

Port B of the deboost valve was pressurized to 1445 psi (the pressure at Port A was atmospheric and the piston was bottomed at the small end). The pressure was held for 2 minutes and then reduced to atmospheric. The test was performed at ambient temperature only.

No fluid leakage was observed during or after the test.

D.3.5 DYNAMIC LEAKAGE, TEST 5

The dynamic leakage test was performed to measure the fluid leakage past the piston seals under dynamic pressure conditions.

The deboost valve was subjected to 25 cycles of the application and release of pressure. The pressure at Port B was cycled between 0 and 965 psi as described in the test procedure, see Appendix C.

No fluid leakage was observed during or after the test.

D.3.6 SEAL FRICTION, TEST 6

The seal friction test was performed to measure the seal friction load which must be overcome before piston motion occurs.

The seal friction was measured by determining the net force (magnitude and direction) acting of the deboost piston before and after motion. The test procedure is described in Appendix C. The test was performed at ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit.

The results of the modified deboost valve seal friction test are given in Table D-4. The results show significant variation in the value of measured friction force and the sign of the force. These variations were also noted in standard deboost valve seal friction test results. The reader is referred to Sections 3.3.1 and D.2.6 for a discussion and explanation of these variations. The average friction force (F) was measured to be approximately -90 pounds at ambient temperature, -2 pounds at -65 degrees Fahrenheit and -69 pounds at 160 degrees Fahrenheit.

D.3.7 FLUID SAMPLES

CTFE fluid samples were taken at the start and end of the deboost valve component performance tests. The fluid samples were drawn from the low pressure port (Port B). Further information concerning the fluid samples is given in Appendix H. The fluid taken at the end of testing showed no apparent discoloration (the CTFE fluid is clear and MIL-H-5606 is red) from the MIL-H-5606.

D.4 STANDARD KC-135 BRAKE COMPONENT PERFORMANCE TEST RESULTS

Two standard KC-135 brakes (serial numbers B0238 and B1043) were subjected to component performance testing. Prior to testing, the original brake piston T-seals were replaced with MS 28775-216 O-rings (see Appendix B). The brakes were then examined and installed in the brake component test setup (see Figures D-7 and D-8). The setup was placed in the environmental test chamber, filled with MIL-H-5606 hydraulic fluid and bled in preparation for testing.

Both standard KC-135 brakes met all the performance requirements established in the Component Performance Test Plan. The results of the tests are summarized in Table D-5. A brief description of each test and the results is given in the following paragraphs.

TABLE D-4 SEAL FRICTION, MODIFIED DEBOOST VALVE

TEMPERATURE DEGREES F	RUN	PRESSURE, PSI PA3	PA5	PB3	PB5	FORCE, lbs F1	F2	F
AMBIENT, 55°F	1	1425	1462	462	469	- 59	-6	- 53
	2	1514	1564	496	500	-120	12	-132
	3	1481	1530	484	493	-104	-30	- 73
	4	1480	1524	485	492	-118	-40	- 78
	5	1520	1623	498	520	-120	1	-121
	6	1453	1498	475	482	-104	-22	- 81
						AVERAGE		- 90
-65°F	1	1300	1743	410	560	72	-16	88
	2	1351	1556	428	499	53	- 4	58
	3	1530	1837	490	598	2	-103	106
	4	1490	1663	476	529	15	42	-26
	5	1415	1589	452	505	15	45	-30
	6	1495	1729	481	552	-21	22	-43
	7	1390	1646	447	525	-17	26	-44
	8	1518	1751	491	560	-50	11	-62
	9	1500	1715	485	548	-48	16	-64
	10	1466	1704	469	545	8	10	- 2
						AVERAGE		- 2
160°F	1	1521	1592	501	518	-150	-87	- 63
	2	1520	1595	501	519	-154	-87	- 66
	3	1406	1453	467	475	-181	-104	- 77
	4	1423	1482	473	485	-188	-111	- 76
	5	1464	1513	486	496	-186	-123	- 62
	6	1448	1512	481	496	-187	-127	- 60
	7	1434	1502	478	493	-204	-129	- 74
	8	1465	1507	487	495	-194	-133	- 60
	9	1530	1609	502	522	-129	- 71	- 58
	10	1490	1588	492	515	-160	- 68	- 92
						AVERAGE		- 69

F1 = PA3 X 3.5332 - PB3 X 11.0270
F2 = PA5 X 3.5332 - PB5 X 11.0270
F = (PB5 - PB3) X 11.0270 - (PA5 - PA3) X 3.5332 = F1 - F2

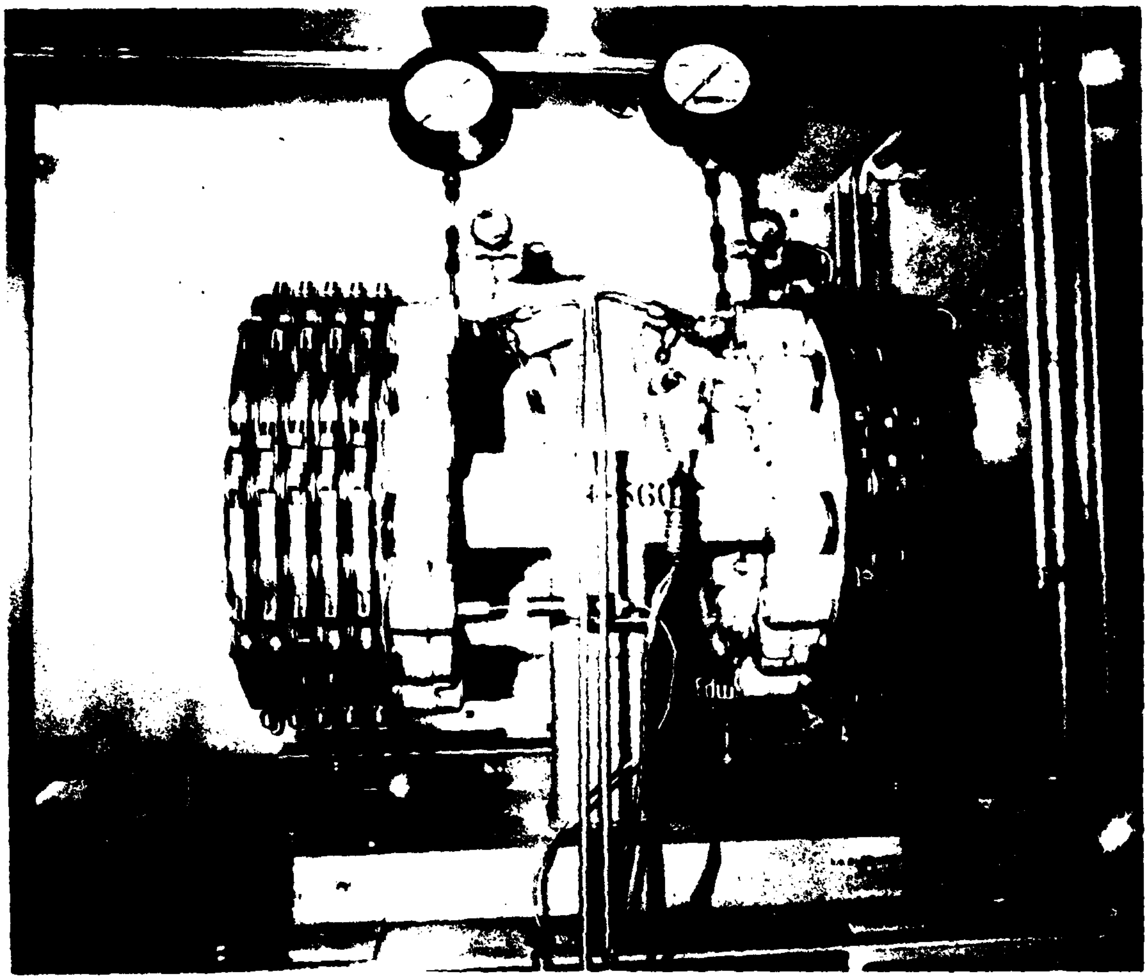


Figure D-7 KC-135 Brake Test Setup

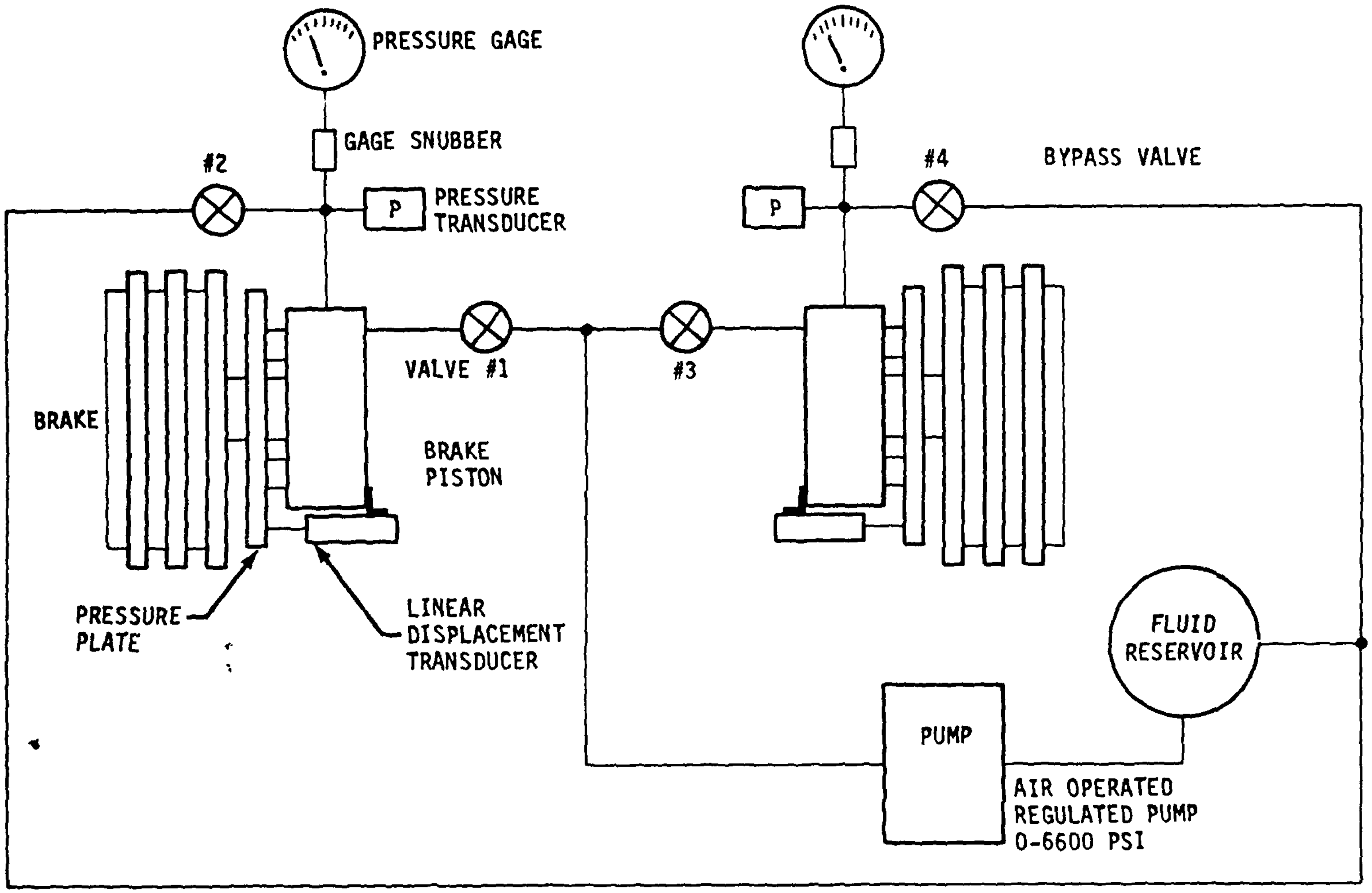


Figure D-8 Brake Test Setup Schematic

TABLE D-5 STANDARD KC-135 BRAKE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEST CONDITIONS TEMP.	RESULTS AND COMMENTS
1. Examination Of Product	—	—	Replaced T-Seal With O-Ring No Wear on Piston or Bushings
2. Seal Break In	50 Cycles,0-300 psi 50 Cycles,0-600 psi 100 Cycles,0-900 psi	Ambient	No Leakage
3. Proof Pressure Static Leakage	1800 psi For 5 Minutes	Ambient	No Leakage
4. Dynamic Leakage			
4a Ambient	25 Cycles,0-1200 psi	Ambient	No Leakage
4b -65°F		-65°F	No Leakage Slow Piston Release
4c 160°F		160°F	No Leakage

The ambient temperature tests were performed on February 9, 1981. The temperature in the test area was 69 degrees Fahrenheit.

The low temperature tests were performed on February 10, 1981. The brakes were soaked in the environmental chamber for approximately 6 hours and 15 minutes at -65 degrees Fahrenheit prior to starting the tests.

The high temperature tests were performed on February 11, 1981. The brakes were soaked in the environmental chamber for approximately 7 hours and 30 minutes at 160 degrees Fahrenheit prior to starting the tests.

D.4.1 EXAMINATION OF PRODUCT, TEST 1

Each brake was visually examined during the seal replacement and prior to component testing. The mating surfaces of the brake pistons and piston bushings were closely inspected. The surface of each part was clean, free of abrasions and free of signs of wear.

Each brake assembly was inspected after the brakes were reassembled and filled with MIL-H-5606 hydraulic fluid. The brakes appeared to be in good condition. No damage was noted or points of fluid leakage observed.

D.4.2 SEAL BREAK IN, TEST 2

The seal break in test was performed to assure proper seating of the brake piston O-ring seals. The seals were seated by cycling brake pressure to move the brake pistons back and forth. Each brake was subjected to 200 cycles of the application and release of pressure as described in the test procedurue. The test was conducted at ambient temperature.

Brake pressure and the brake pressure plate displacement were monitored during the tests. The data indicated that each brake piston was displaced over its full range of motion (approximately 0.23 inches) during each pressure cycle.

The brakes were examined several times during and at the end of testing. No fluid leakage was observed.

D.4.3 PROOF PRESSURE AND STATIC LEAKAGE, TEST 3

The proof pressure and static leakage test was performed to measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under static conditions a proof pressure.

Each brake was pressurized to 1800 psi for 5 minutes. The brake pressure was then relieved (atmospheric). The test was performed at ambient temperature.

During the test, brake pressure and brake pressure plate displacement were monitored. The brake pressure was observed to sag approximately 40 to 60 psi during the first 2 1/2 minutes of the test. The pressure then held constant at approximately 1750 psi for the duration of the test. A time history plot of brake pressure is shown in Figure D-9.

The observed sag in pressure is due to an incremental compression of the brake stack which occurs as the rotors and stators continue to seat against one another during the initial 2 1/2 minutes. The stack compression causes the hydraulic volume to expand slightly thereby reducing brake pressure. The change in fluid volume caused by a 50 psi change in brake pressure is estimated to be 0.0098 cubic inch. This corresponds to a increase in the brake piston displacement of 0.0008 inch. The smallest change in piston displacement which could be detected with the instrumentation was approximately .0016 inch. Thus, the change in brake piston displacement was not detected or observed on the recorded data.

When the brake pressure was relieved the brake pistons returned to the retracted position (as determined by brake pressure plate displacement). No permanent set or change in the retracted position was observed.

No fluid leakage was observed during or after the test.

D.4.4 DYNAMIC LEAKAGE, TEST 4

The dynamic leakage test was performed to measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under dynamic pressure conditions.

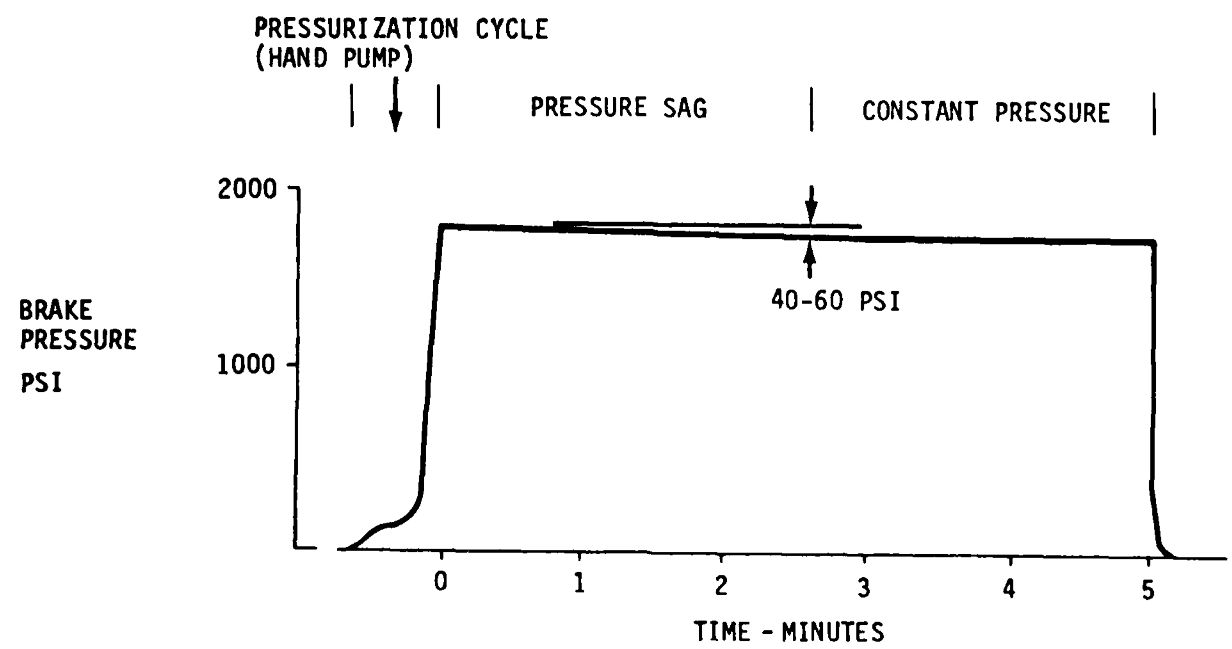


Figure D-9 Brake Pressure Time History

Each brake was subjected to 25 cycles of the application and release of 1200 psi. The brake pistons were allowed to return to the fully retracted or equilibrium position (as determined by the displacement of the brake pressure plate) after each release of pressure (0 psi) and prior to reapplication of pressure. The test was performed at ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit.

The time required to dump brake pressure and the time for the brake pistons to reach an equilibrium position was measured at each test temperature. Typical time history plots of brake pressure plate displacement and brake pressure for the two brakes are shown in Figures D-10 and D-11. Brake pressure was released by opening the bypass valve (valve #2 or #4 in Figure D-8) one full turn (manually opened, one quick motion). Some difference in the time required for the brake piston to reach the retracted position was observed between the two brakes. This variation in time is due to slight differences between the right and left brake test set up.

The brakes were examined during and after the test. No fluid leakage was observed.

D.5 MODIFIED BRAKE COMPONENT PERFORMANCE TEST RESULTS

Two KC-135 brakes (serial numbers B0145 and B1212) modified for use with CTFE hydraulic fluid were tested during the brake performance component tests. The brakes were modified as described in Appendix B prior to the testing. The brakes were examined during the modification process to assure that they were cleaned and reassembled properly. The brakes were then installed in the brake component test setup (see Figures D-12 and D-8). The test setup was placed in the environmental chamber, filled with CTFE hydraulic fluid and bleed prior to testing.

The modified brakes met all the component performance requirements established in the test plan. The results of the component tests are summarized in Table D-6. A brief description of each test and the results is given in the following paragraphs.

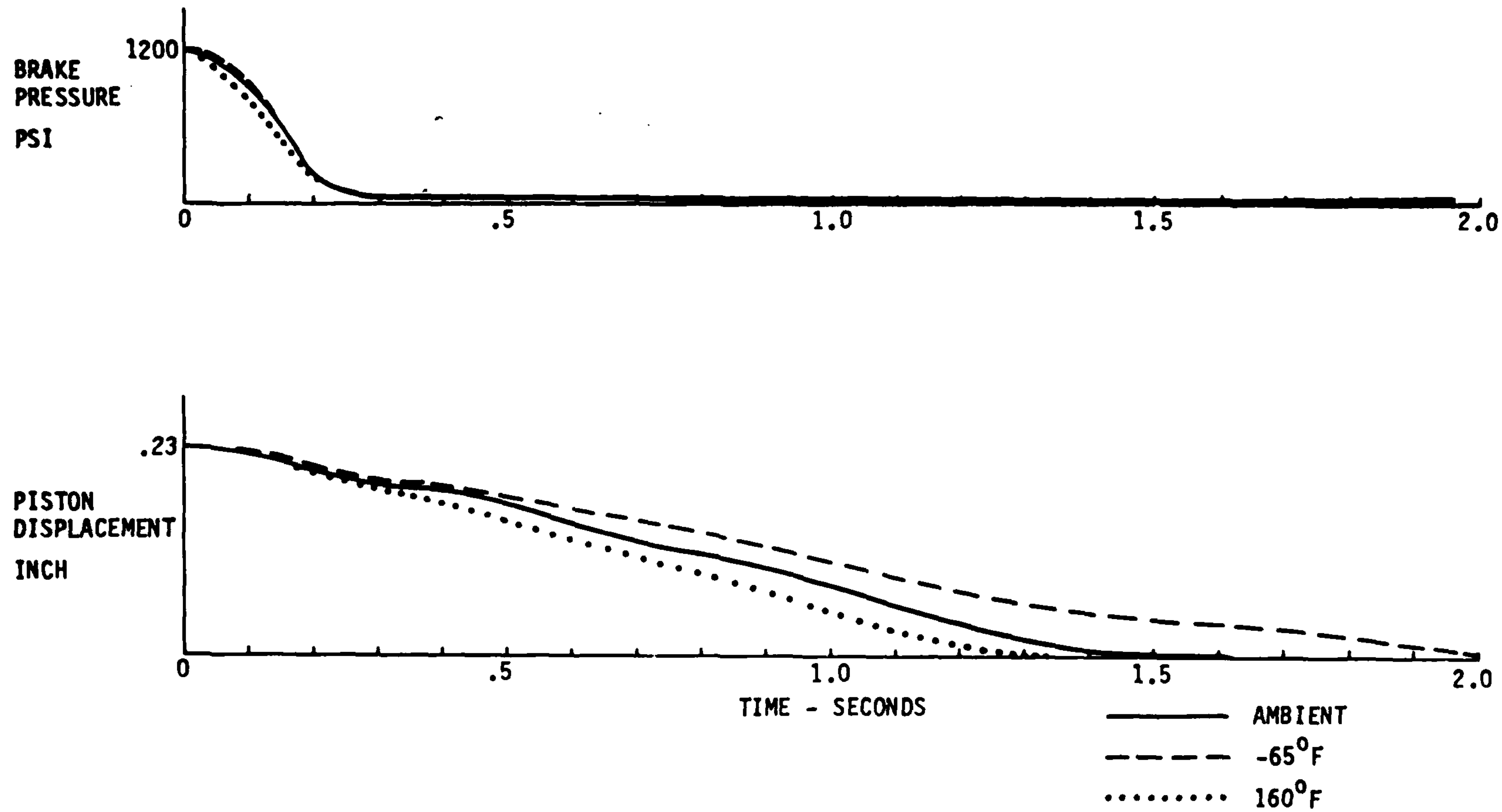


Figure D-10 Brake Pressure and Piston Displacement, Brake SN B0238

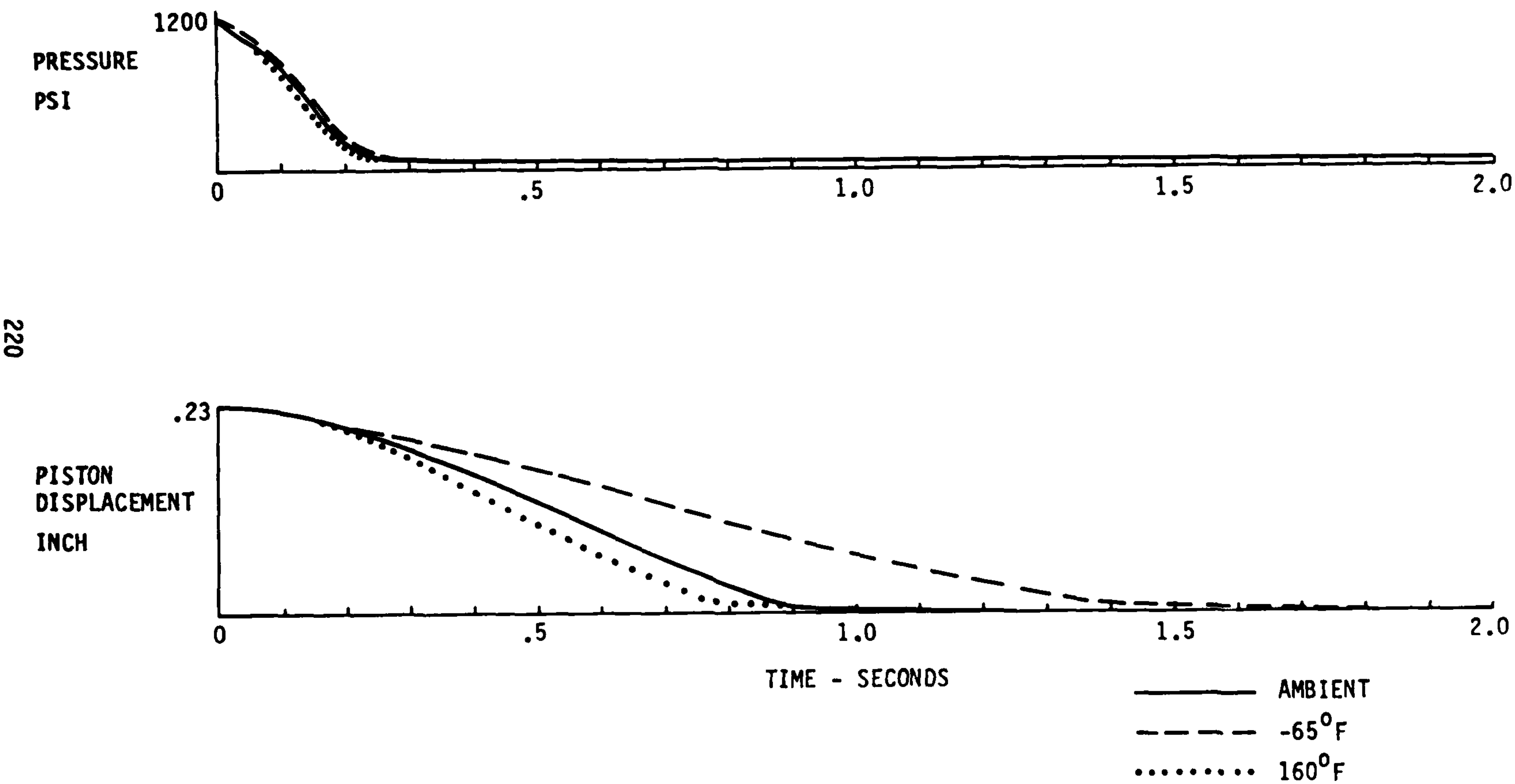


Figure D-11 Brake Pressue and Piston Displacement, Brake SN B1043

TABLE D-6 MODIFIED BRAKE COMPONENT PERFORMANCE TEST RESULTS

TEST	TEST CONDITIONS DESCRIPTION	TEMP.	RESULTS AND COMMENTS
1. Examination Of Product	—	—	Cleaned and Inspected all Part, Replaced T-Seal With PNF O-Rings
2. Seal Break In	50 Cycles,0-300 psi 50 Cycles,0-600 psi 100 Cycles,0-900 psi	Ambient	No Leakage
3. Proof Pressure Static Leakage	1800 psi For 5 Minutes	Ambient	No Leakage
4. Dynamic Leakage			
4a Ambient	25 Cycles,0-1200 psi	Ambient	No Leakage
4b -65°F		-65°F	No Leakage Slow Piston Release
4c 160°F		160°F	No Leakage

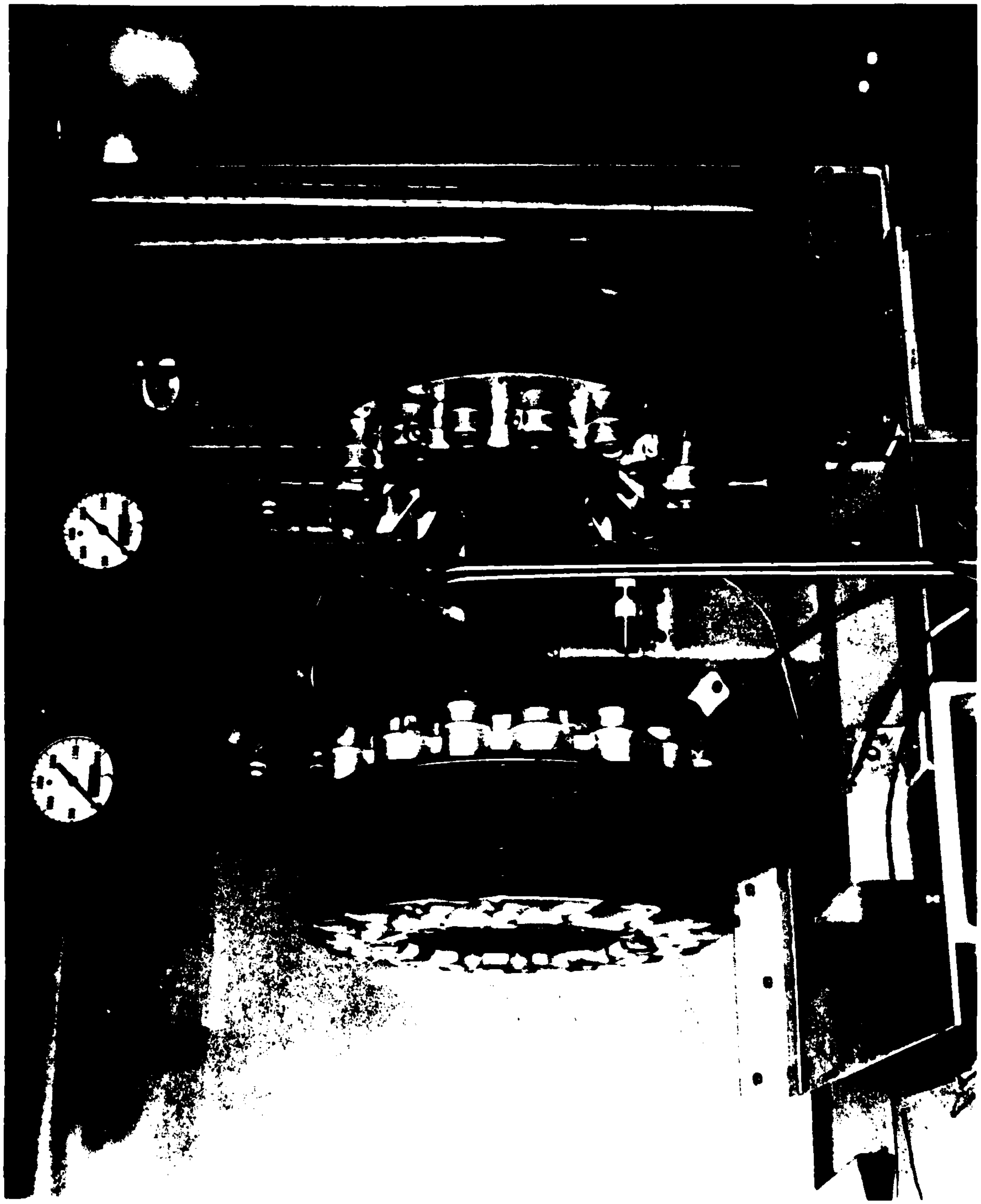


Figure D-12 Modified Brake Test Setup

The ambient temperature tests were performed on February 13, 1981. The temperature in the test area was 67 degrees Fahrenheit.

The low temperature tests were performed on February 16, 1981. The brakes were soaked in the environmental chamber for approximately 6 hour and 30 minutes at -65 degrees Fahrenheit prior to the start of testing.

The high temperature tests were performed on February 17, 1981. The brakes were soaked in the environmental chamber for approximately 7 hours and 10 minutes at 160 degrees Fahrenheit prior to the start of testing.

D.5.1 EXAMINATION OF PRODUCT, TEST 1

Each brake was visually examined during the modification procedure and prior to testing. During the modification the cleaned brake housing, new pistons and new piston bushings were carefully examined. Each part was clean, free of scratches and free of wear. The brakes were reassembled using PNF O-ring seals. The brakes were then placed in the test setup, filled with CTFE hydraulic fluid and inspected. No signs of fluid leakage were observed during the inspection.

D.5.2 SEAL BREAK IN, TEST 2

The seal break in test was performed to assure proper seating of the PNF seals. The seals were seated by cycling brake pressure to move the brake pistons back and forth. Each brake was subjected to 200 cycles of the application and release of pressure as described in the test procedure. The test was conducted at ambient temperature.

Brake pressure and the brake pressure plate displacement were monitored during the test. The data indicated that the brake pistons were displaced approximately 0.23 inches during each pressure cycle.

Each brake was examined several times during and at the end of testing. No fluid leakage was observed.

D.5.3 PROOF PRESSURE AND STATIC LEAKAGE, TEST 3

The proof pressure and static leakage test was performed to measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under static conditions at proof pressure.

Each brake was pressurized to 1800 psi for 5 minutes. The brake pressure was then relieved (atmospheric). The test was performed at ambient temperature.

The brake pressure and brake pressure plate displacement were monitored during the test. The brake pressure was observed to sag about 40-60 psi immediately after application of pressure. This sag was also noted in the MIL-H-5606 fluid brake tests and is due to continued seating of the brake rotors and stators. Several additional test repetitions were performed for verification. During these runs the brake stack was tapped lightly with a hammer during pressurization. The induced vibration tended to seat the rotors and stators against one another during stack compression. The sag in pressure was not observed during these runs.

When brake pressure was relieved the brake pistons returned to the retracted position. No permanent set or change in the equilibrium retracted position was noted.

No fluid leakage was observed during or after the test.

D.5.4 DYNAMIC LEAKAGE, TEST 4

The dynamic leakage test was performed to measure the fluid leakage at static and dynamic seals and observe the apparent seal friction under dynamic pressure conditions.

Each brake was subjected to 25 cycles of the application and release of 1200 psi. The brake pistons were allowed to fully retract (reach an equilibrium position) after each release of pressure and before reapplication of pressure. The test was performed at ambient, -65 degrees Fahrenheit and 160 degrees Fahrenheit.

The time required to dump brake pressure and the time for the brake pistons to reach the retracted position was measured at each test temperature. Typical time history plots of brake pressure and brake pressure plate displacement are shown in Figures D-13 and D-14. Brake pressure was dumped by rapidly opening the bypass valve #2 or #4 (Figure D-8) one full turn. Some difference in the piston retraction time was noted between the two brakes. This same difference was also observed in the MIL-H-5606 fluid brake tests. Comparison of the test results indicates that the difference is due to slight differences between the right and left brake test setup.

The brakes were examined after each test. No fluid leakage was observed.

D.5.5. FLUID SAMPLES

CTFE fluid samples were taken from the modified brake at the start and end of the component performance tests. The fluid samples were supplied to AFWAL/MLBT for analysis. Further information concerning the fluid samples is given in Appendix H.

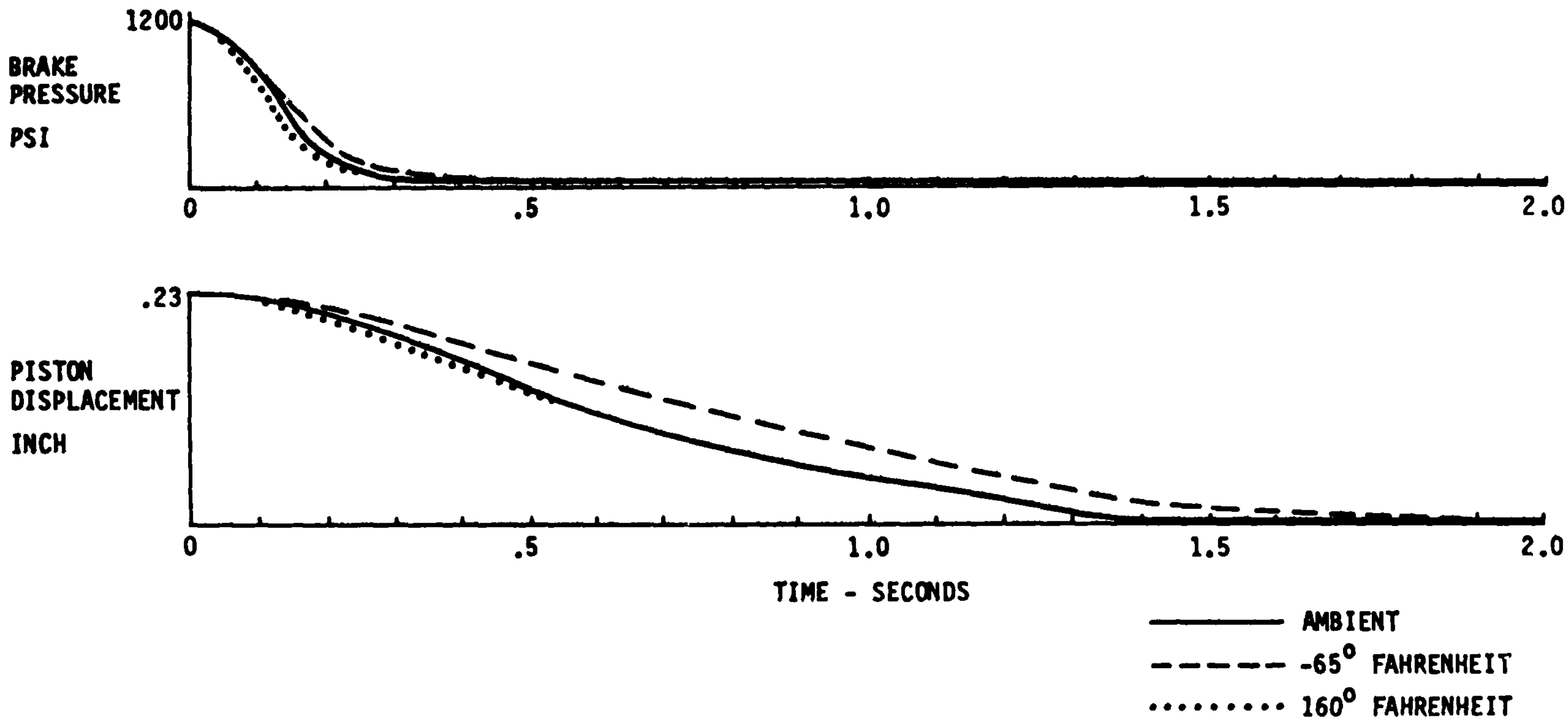


Figure D-13 Brake Pressure and Piston Displacement, Brake SN B0145

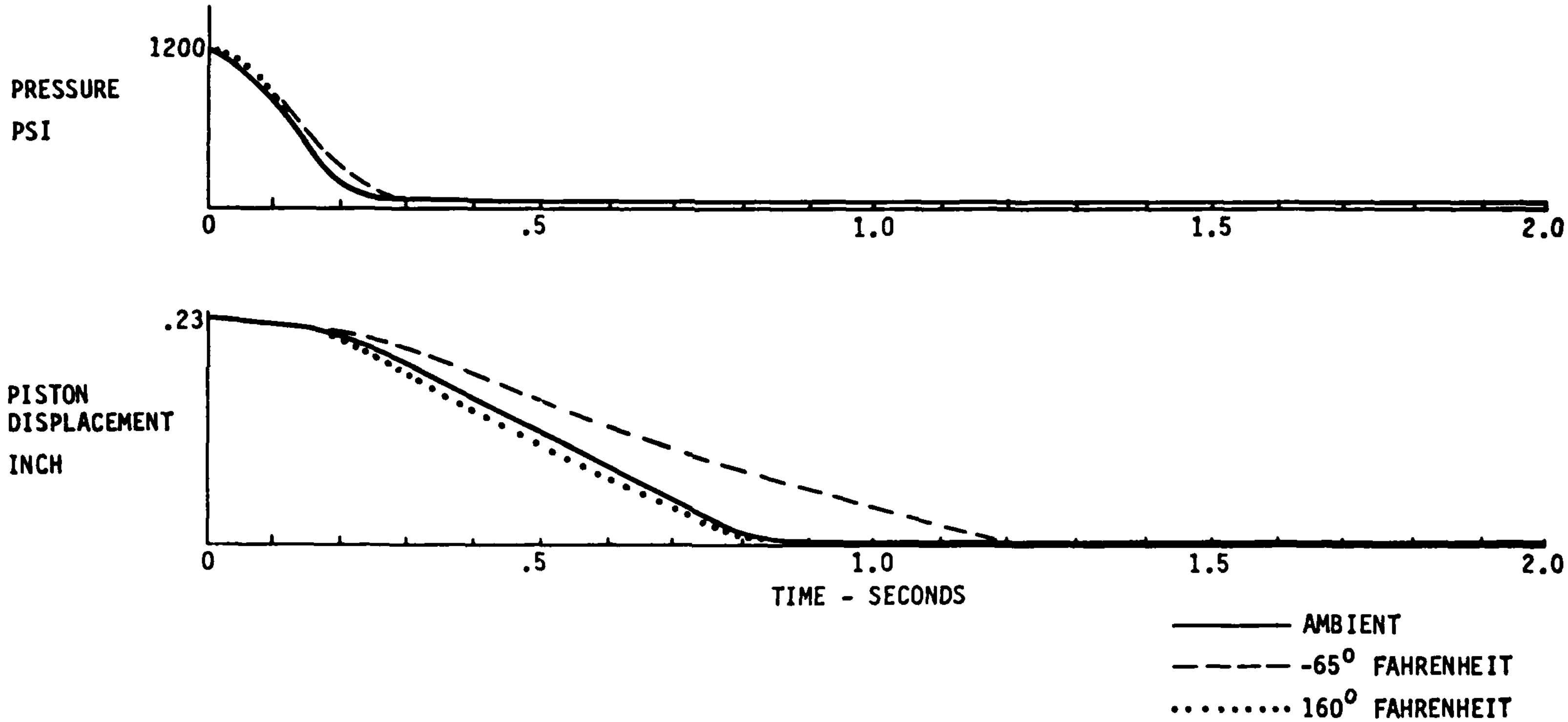


Figure D-14 Brake Pressure and Piston Displacement, Brake SN B1212

APPENDIX E

SYSTEM PERFORMANCE TEST RESULTS

E.1 STANDARD KC-135 BRAKE HYDRAULIC SYSTEM PERFORMANCE TEST RESULTS

System performance tests were conducted using the KC-135 brake hydraulic system mockup shown in Figure 18. A schematic of the system mockup, Figure E-1 and Table E-1 are included to define the configuration of the test setup and instrumentation points. The unmodified KC-135 deboost valve and brakes tested during the component performance testing were used in the mockup. The components used in the mockup are listed in Table E-2. All the components are standard KC-135 parts and were supplied by the Air Force as GFP. The mockup constructed with these component duplicated the actual KC-135 brake system configuration. Line diameters, lengths and tubing wall thickness were duplicated. Additional fittings not found in the actual system were included for necessary instrumentation. The instrumentation points are defined in Figure E-1.

The system mockup was placed in the environmental chamber (shown in Figure D-1) and serviced with MIL-H-5606 hydraulic fluid prior to testing. The mockup was integrated with the KC-135 Mark II antiskid control unit and the hybrid computer to form the KC-135 airplane simulation.

The system tests were performed at ambient, +160 degrees Fahrenheit and -40 degrees Fahrenheit. The full series of system tests were performed at these temperatures. The original test plan called for the low temperature tests to be performed at -65 degrees Fahrenheit. Approximately 90% of the system tests were completed (Test 2, Step Response was not performed) at -65 degrees Fahrenheit when it was noticed that fluid was leaking from the dynamic seals in the brakes and from static seals between the first and second stages of the antiskid valve. Judging from the large quantity of fluid in the test chamber and the observed flow rate, the leakage apparently started midway through the testing. There is no indication in the test results of when the leakage started although system performance may have been affected. The -65 degrees

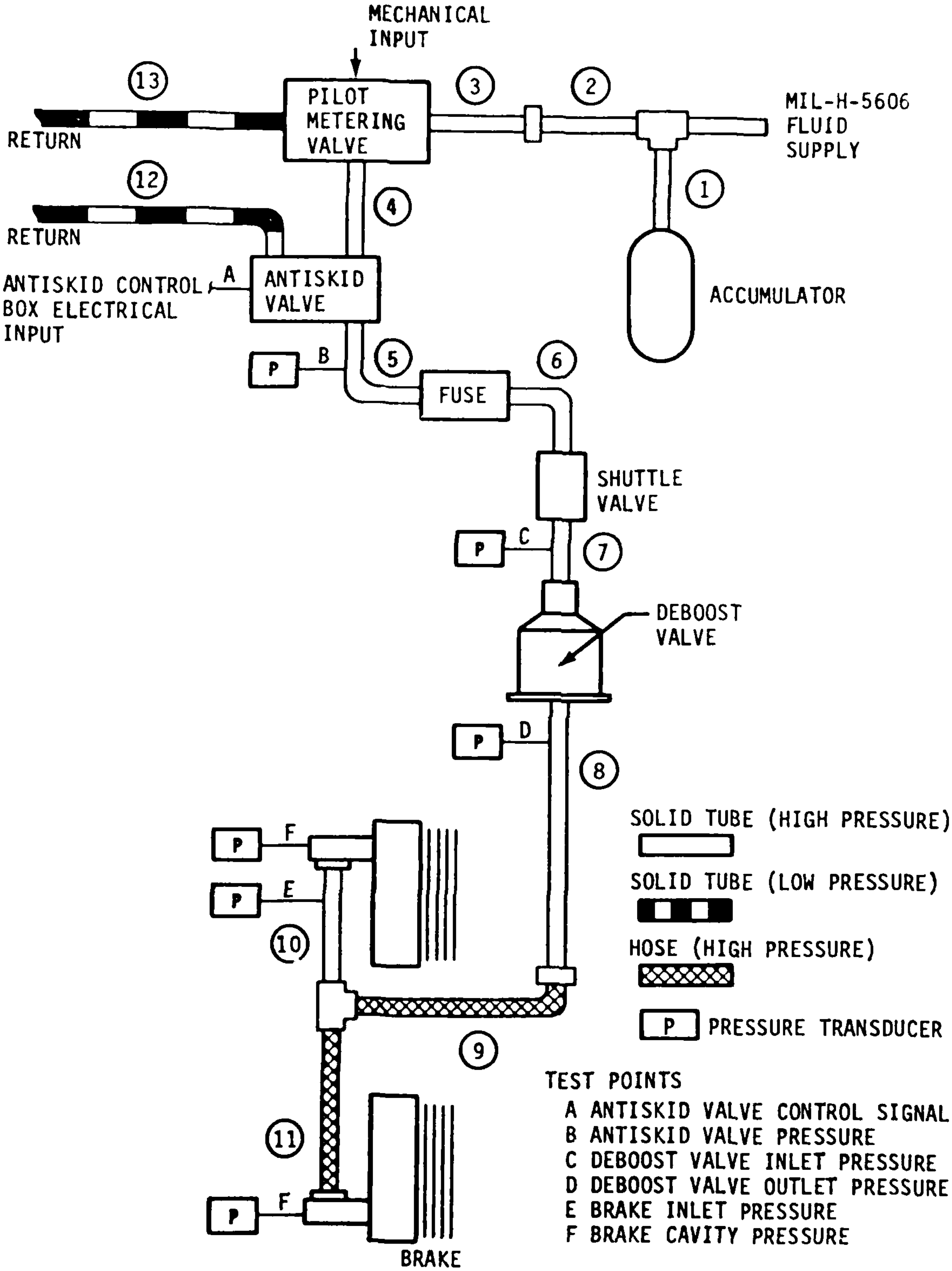


Figure E-1 KC-135 Brake Hydraulic System Mockup Schematic

TABLE E-1 KC-135 BRAKE HYDRAULIC SYSTEM MOCKUP DATA

DESCRIPTION	LINE NUMBER	LINE SIZE[3]	LINE LENGTH (Inches)
ACCUMULATOR LINE	1	6S35	60
SUPPLY PRESSURE LINE	2	8S49	60
	3	6S35	60
METERED PRESSURE LINE	4	8S49	60
BRAKE PRESSURE LINE	5	8S49	11
	6	8S49	16
	7 [1]	8S49	5
	8	8S49	170
	9	1/2 inch hose[2]	81
BRAKE #1 PRESSURE LINE	10	6S35	63
BRAKE #2 PRESSURE LINE	11	3/8 inch hose[2]	24
PILOT METERING VALVE RETURN	12	8A35	AS REQUIRED
ANTISKID VALVE RETURN	13	8A35	AS REQUIRED

1 THIS LINE NOT PRESENT IN ACTUAL KC-135 BRAKE HYDRAULIC SYSTEM. IT WAS REQUIRED FOR INSTRUMENTATION OF DEBOOST VALVE

2 HIGH PRESSURE, TEFLON LINED, STAINLESS STEEL JACKET HOSES

3 TUBING DESIGNATION - 8S49

- WALL THICKNESS IN THOUSANDTHS
- STAINLESS STEEL (S) OR ALUMINUM (A)
- O.D. IN 1/16THS

TABLE E-2 KC-135 BRAKE HYDRAULIC SYSTEM MOCKUP COMPONENTS

ITEM	NATIONAL STOCK NUMBER	QUANTITY
SKID CONTROL BOX (NOT SHOWN IN FIGURE E-1)	1630-00-918-0340	1
PILOT METERING VALVE	1630-00-610-7199	1
DUAL ANTISKID VALVE	1630-00-908-9999	1
FUSE	1650-00-672-8013	1
DEBOOST VALVE	1650-00-570-8397	1
ACCUMULATOR	1650-00-584-9343	1
BRAKE ASSEMBLY	1630-00-058-5242	2

Fahrenheit tests were discontinued after the leakage was noticed. A short series of tests were performed at ambient and +160 degrees Fahrenheit to determine the extent of the seal leakage problem. No leakage was observed from either the brakes or the antiskid valve during these tests. The leakage problem is apparently temperature related and due to poor sealing capability at -65 degrees Fahrenheit. The validity of the test results is questionable due to the unknown effect which leakage has upon system performance. Although the data is questionable and incomplete it is included here for reference.

The low temperature test condition was raised to -40 degrees Fahrenheit and the system tests rerun. Minor leakage from the antiskid valve occurred for approximately one minute after initial pressurization of the system. No further leakage occurred during the duration of the testing. The decision to raise the test temperature to -40 degrees F and continue testing was made with the concurrence of the Air Force Project Engineer.

The ambient temperature tests were performed on April 2, 1981. The temperature in the test area was 73 degrees Fahrenheit.

The high temperature tests were performed on April 3, 1981. The brake hydraulic system was soaked for 6 hours and 10 minutes at 160 degrees Fahrenheit prior to the start of testing.

The abbreviated series of low temperature tests at -65 degrees Fahrenheit were performed on April 7, 1981. The system was soaked approximately 6 hours and 25 minutes prior to the start of testing.

The low temperature tests at -40 degrees Fahrenheit were performed on April 10, 1981. The brake hydraulic system was soaked about 6 hours and 30 minutes at -40 degrees Fahrenheit prior to the start of testing.

E.1.1 FREQUENCY RESPONSE, TEST 1

Frequency response tests were performed to determine the dynamic response of the hydraulic system, deboost valve and antiskid valve to a sinusoidal antiskid valve control signal.

A D.C. electrical control signal corresponding to the desired D.C. pressure level was applied to the antiskid valve. A 0.5 hertz sinusoidal electronic signal was superimposed on the D.C. signal. The amplitude of the sinusoidal signal was adjusted until the desired sinusoidal pressure amplitude of the brake was obtained. The frequency of the sinusoidal signal was varied between 0.5 Hertz and 50 Hertz. The dynamic gain and phase angle of the system (Test 1a), antiskid valve (Test 1b.) and deboost valve (Test 1c.) were determined as a function of frequency. The test plan also called for the determination of the frequency response of the brake. Upon examination of the brake it was found that the input (test point E, Figure E-1) and output (test point F) pressure ports used to determine the response were located in the same boss. Consequently, a meaningful brake frequency response could not be determined. The Air Force Project Engineer was informed and the test was deleted from the system test plan with Air Force concurrence.

The frequency response test conditions, and test points are given in Table E-3. The test results are given in Figures E-2 thru E-13.

E.1.2 STEP RESPONSE, TEST 2

The step response tests were performed to determine the dynamic response of the brake hydraulic system or a series of components to a step change in the antiskid valve control signal.

A D.C. electrical control signal corresponding to the initial brake pressure test level was applied to the antiskid valve. The control signal was then stepped up or down to a level corresponding to the final pressure level. The step response test conditions and test points are given in Table E-4. The response of the brake system (at several test points) to a step pressure change command occurring at time zero is shown in Figures E-14 thru E-21.

E.1.3 STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE, TEST 3

The pressure-current characteristic of the antiskid valve was measured to determine whether the valve met the manufacturer's specifications.

TABLE E-3 FREQUENCY RESPONSE TEST CONDITIONS

TEST	TEST TEMPERATURE	TEST POINTS INPUT	TEST POINTS OUTPUT	FREQUENCY RANGE (HZ)	TEST CONDITION PRESSURE MEASURED AT BRAKE
1a. Brake System	Ambient -65°F 160°F	A	F	0.5 - 50	33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi
1b. Antiskid Valve	Ambient -65°F 160°F	A	B		33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi
1c. Deboost Valve	Ambient -65°F 160°F	C	D		33% full pressure ±100 psi 33% full pressure ±200 psi 66% full pressure ±100 psi 66% full pressure ±200 psi

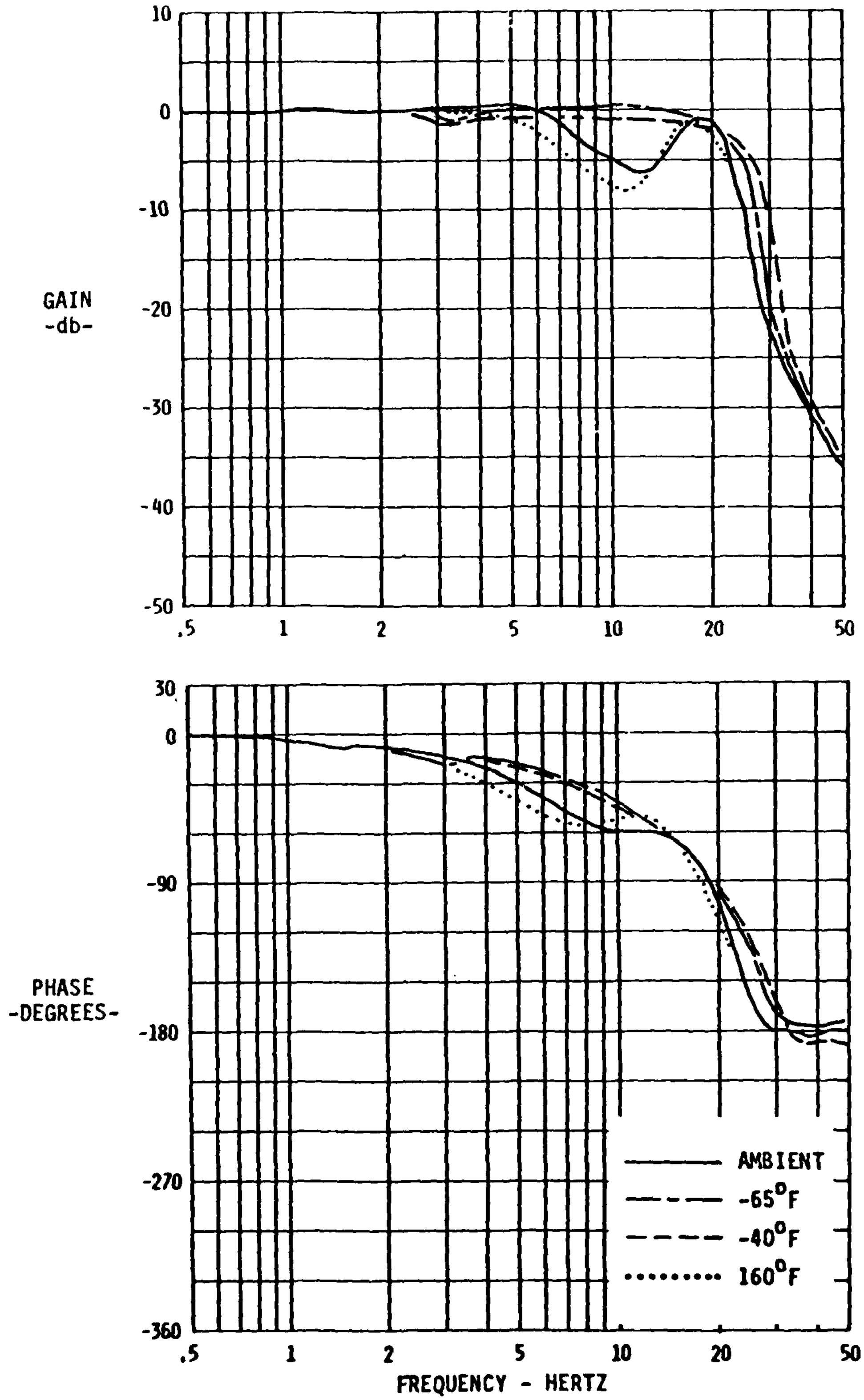


Figure E-2 Frequency Response, Standard Antiskid Valve, 33% ± 100 psi

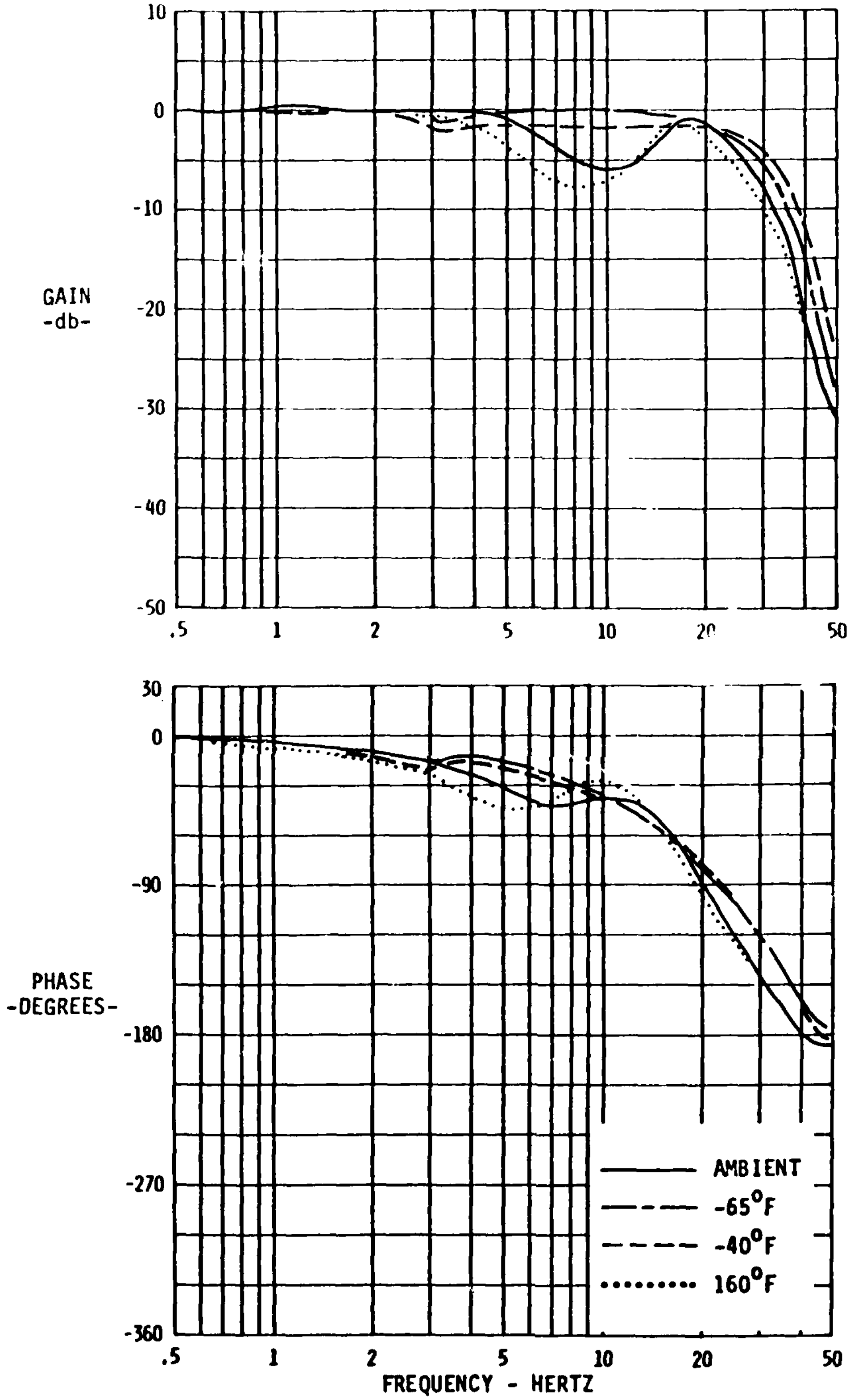


Figure E-3 Frequency Response, Standard Antiskid Valve, 33% ± 200 psi

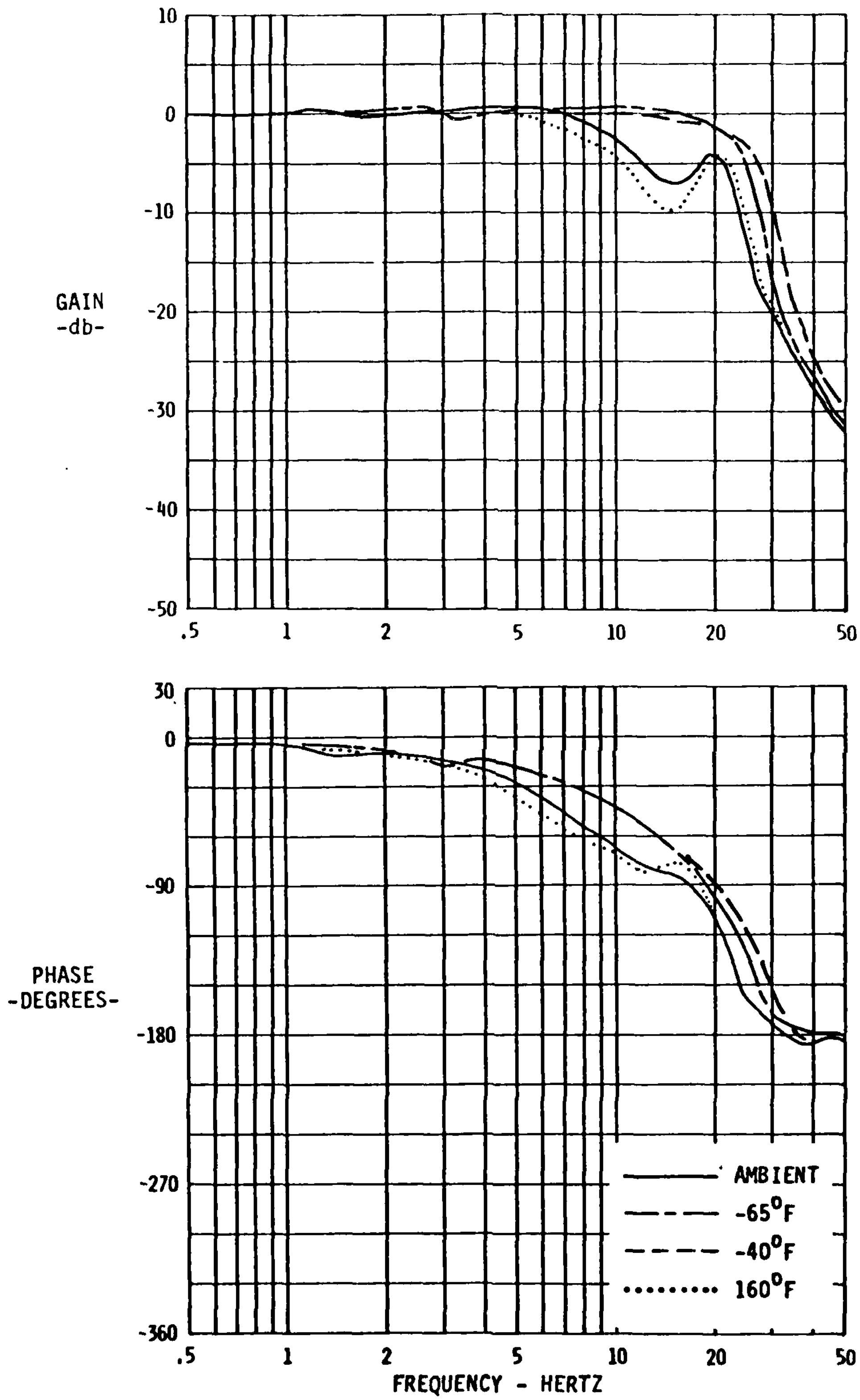


Figure E-4 Frequency Response, Standard Antiskid Valve, 66% ± 100 psi

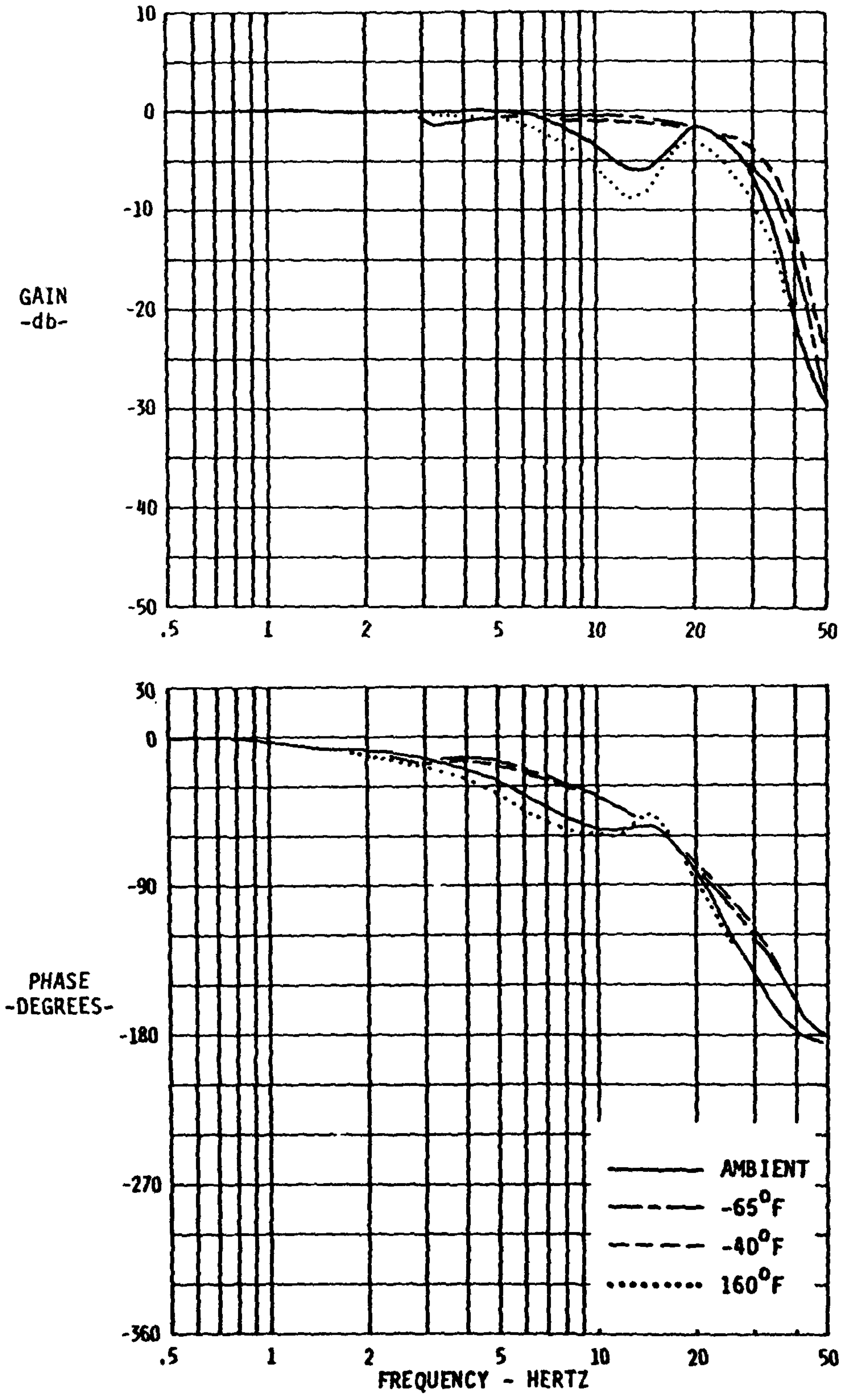


Figure E-5 Frequency Response, Standard Antiskid Valve, 66% ± 200 psi

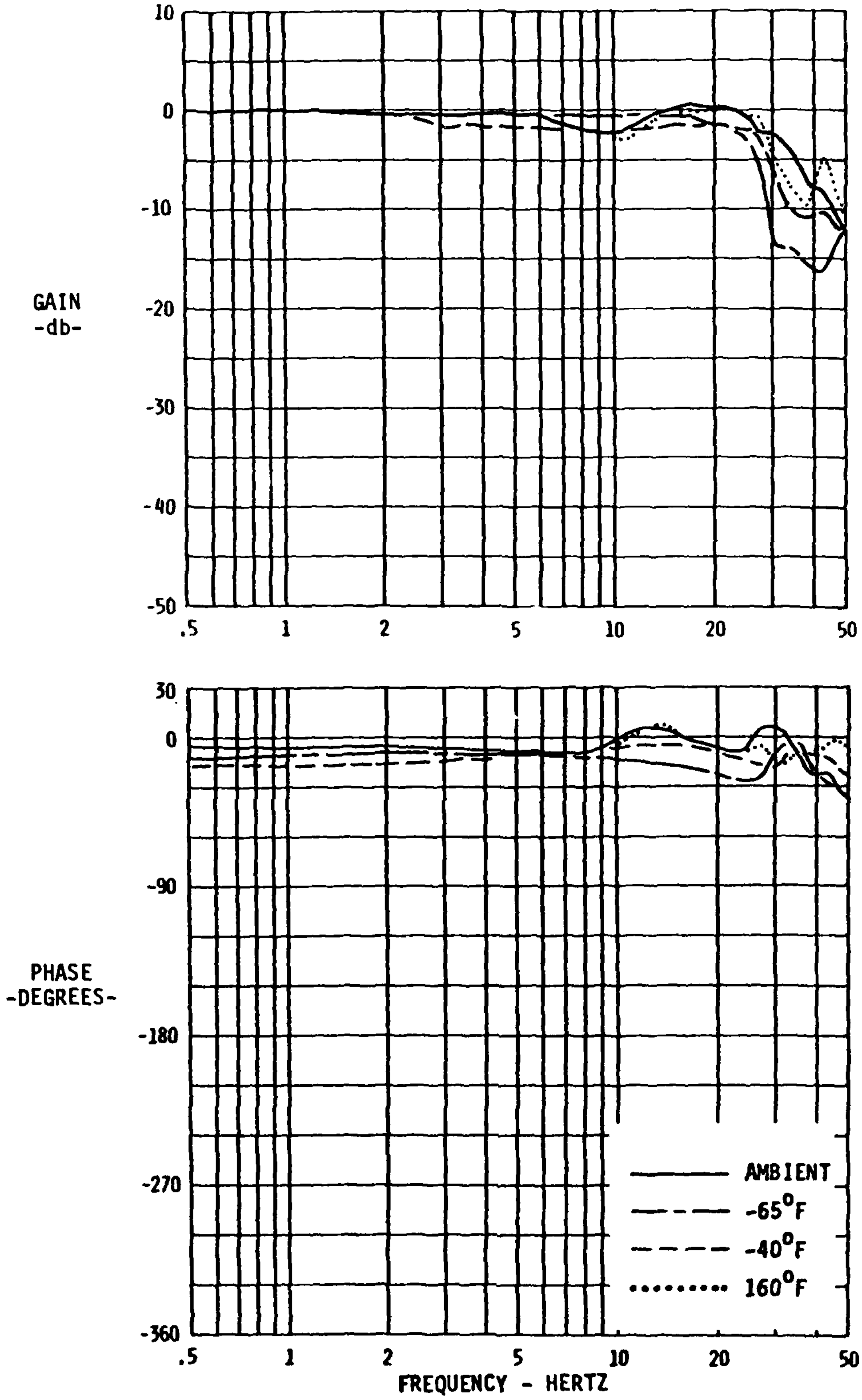


Figure E-6 Frequency Response, Standard Deboost Valve, 33% ± 100 psi

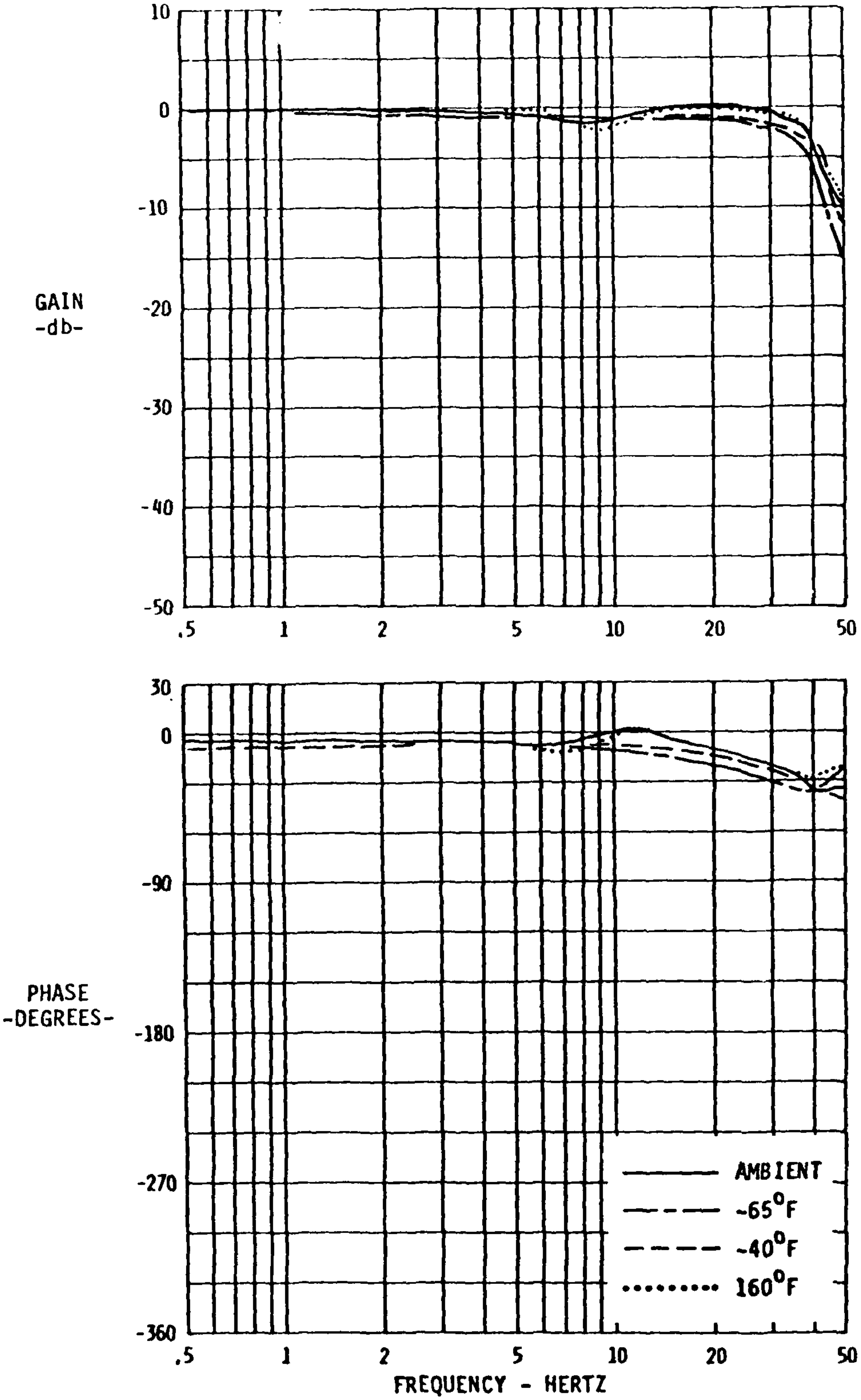


Figure E-7 Frequency Response, Standard Deboost Valve, 33% ± 200 psi

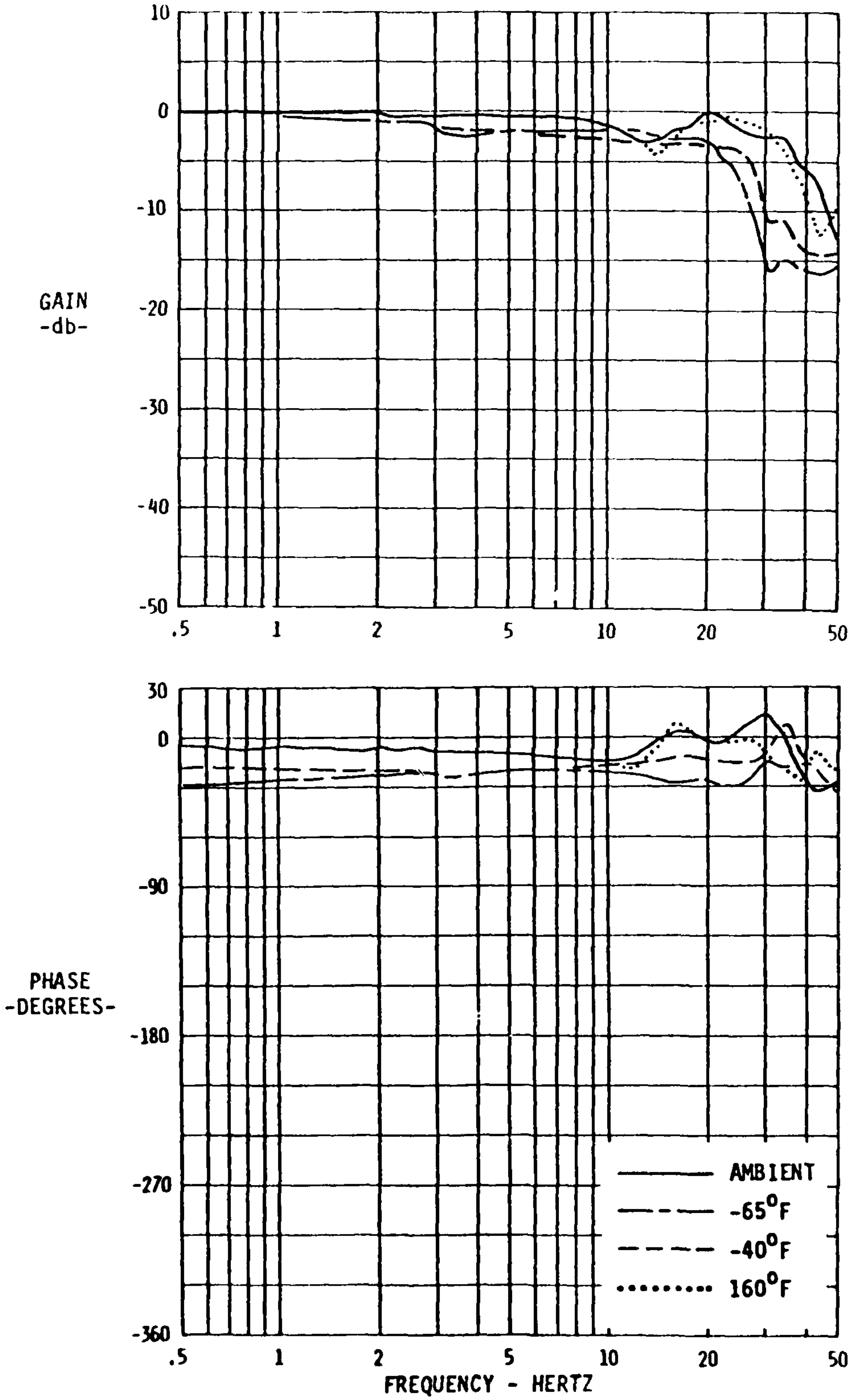


Figure E-8 Frequency Response, Standard Deboost Valve, 66% ± 100 psi

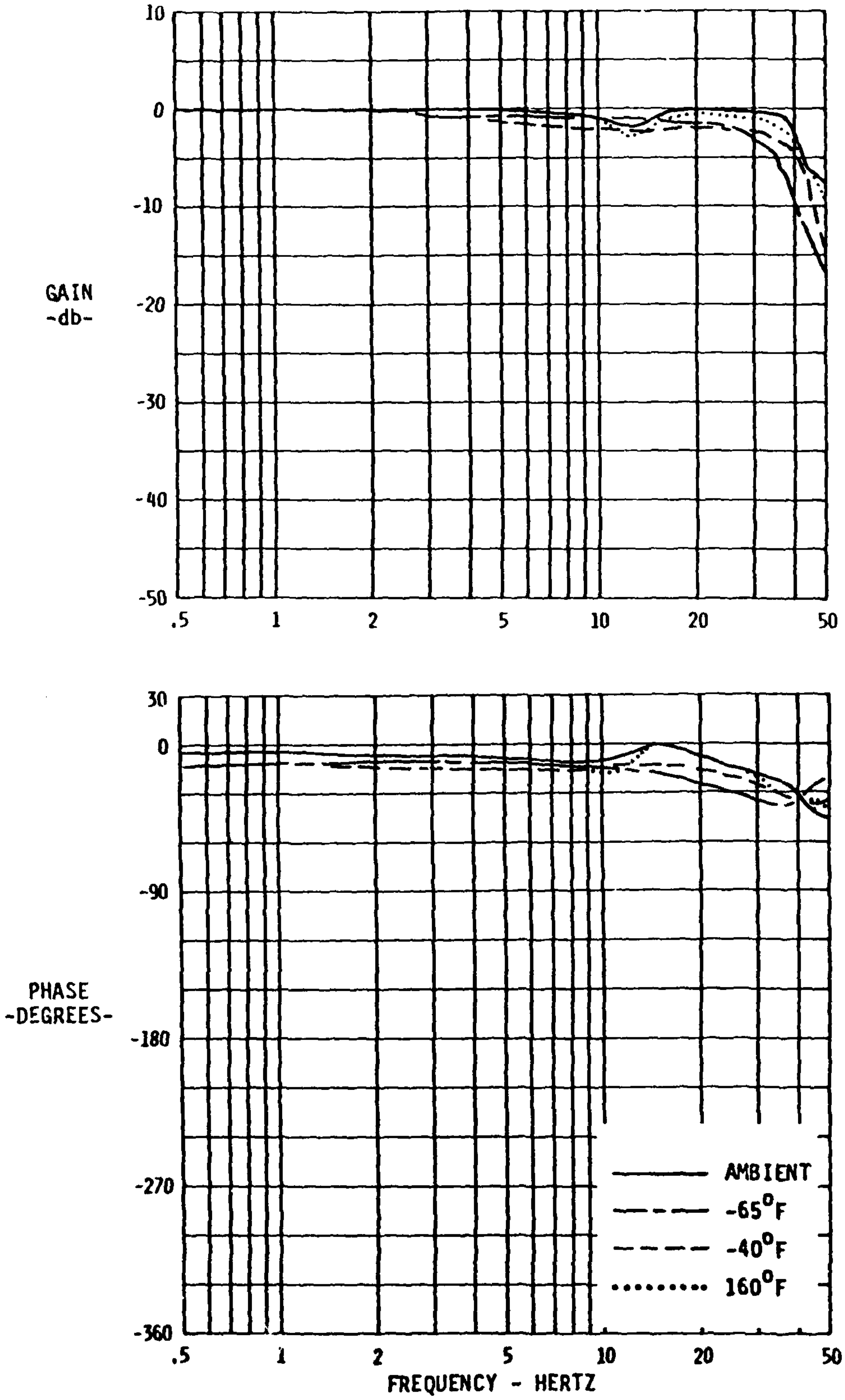


Figure E-9 Frequency Response, Standard Deboost Valve, 66% ± 200 psi

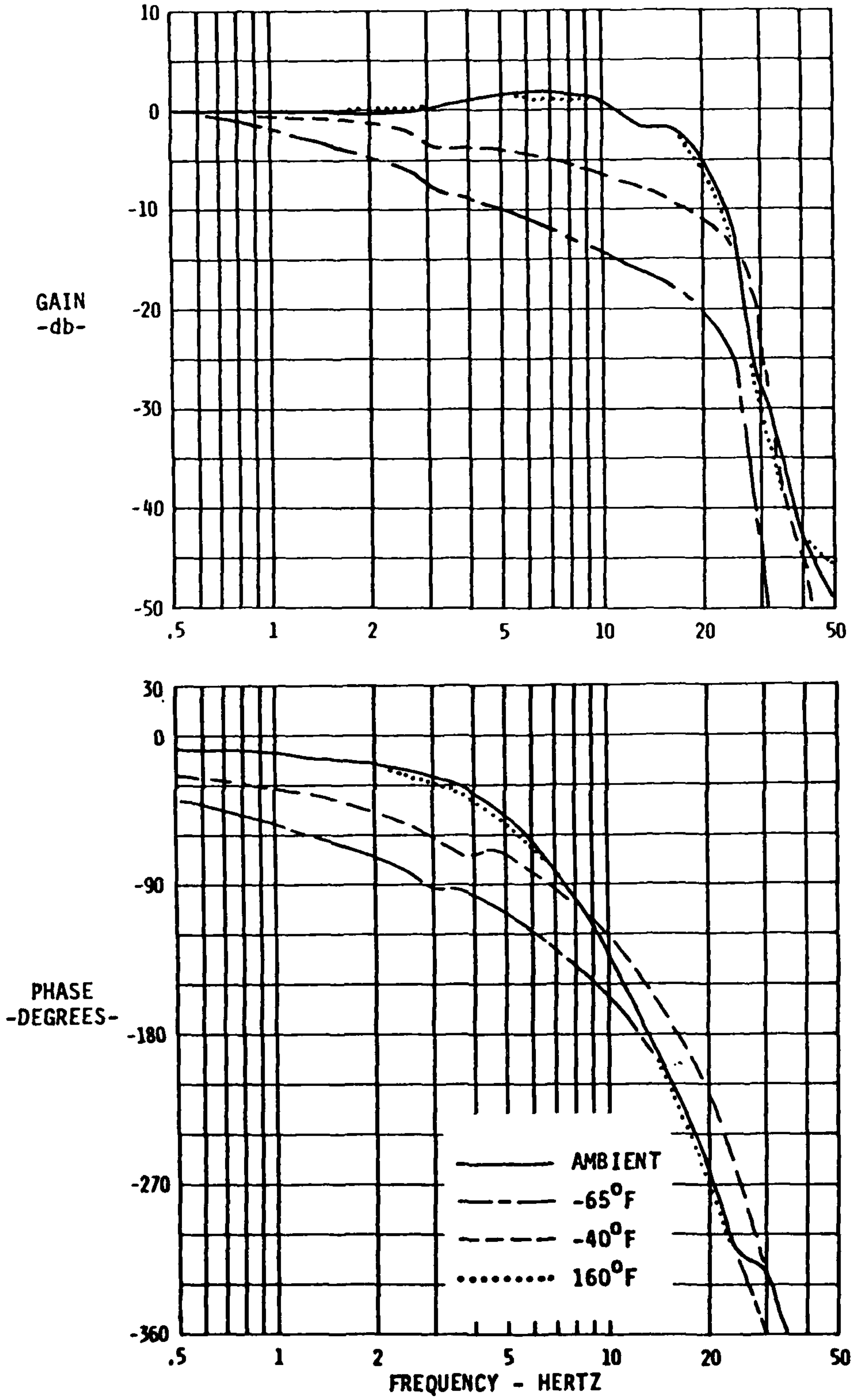


Figure E-10 Frequency Response, Standard Brake System, 33% ± 100 psi

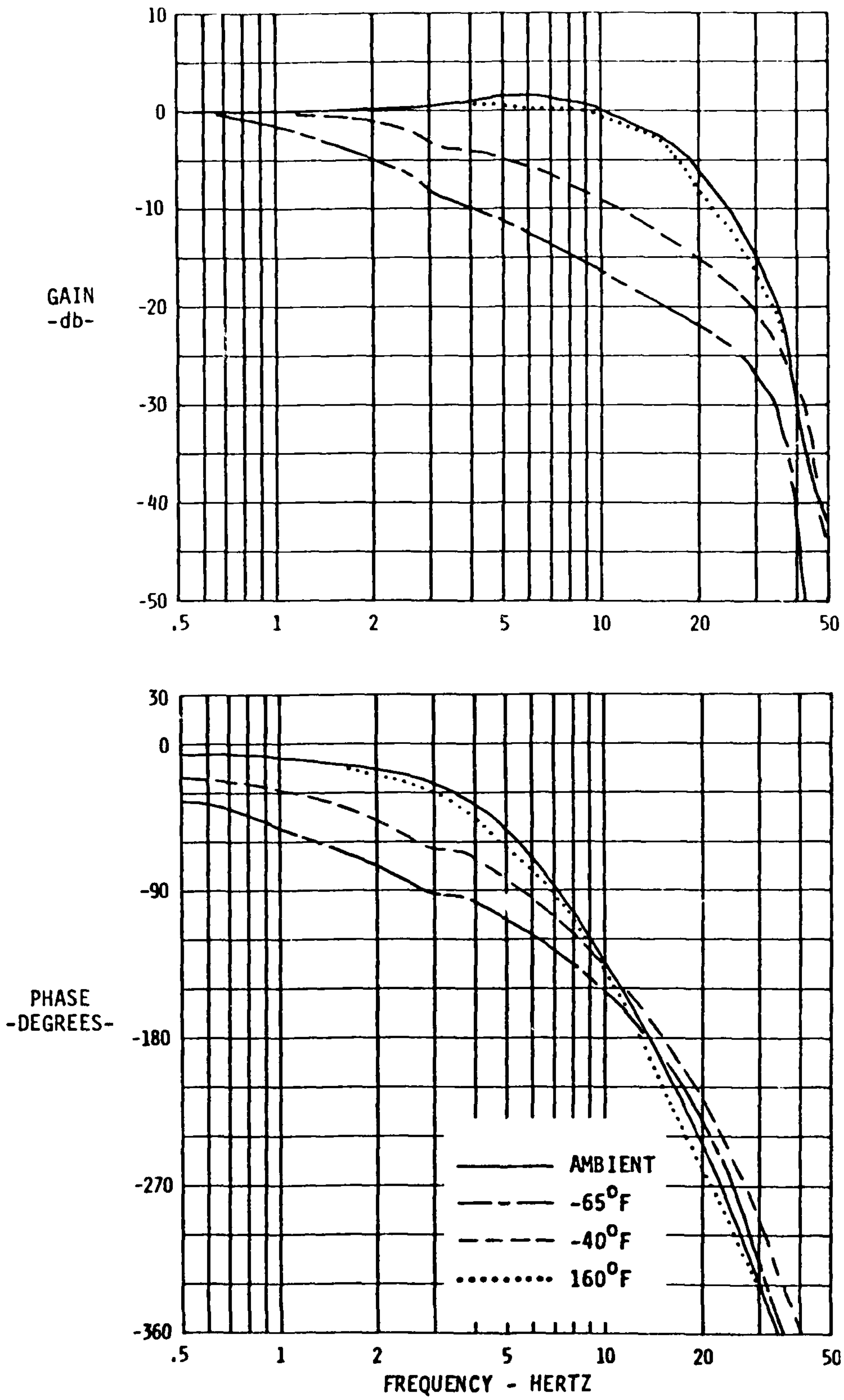


Figure E-11 Frequency Response, Standard Brake System, 33% ± 200 psi

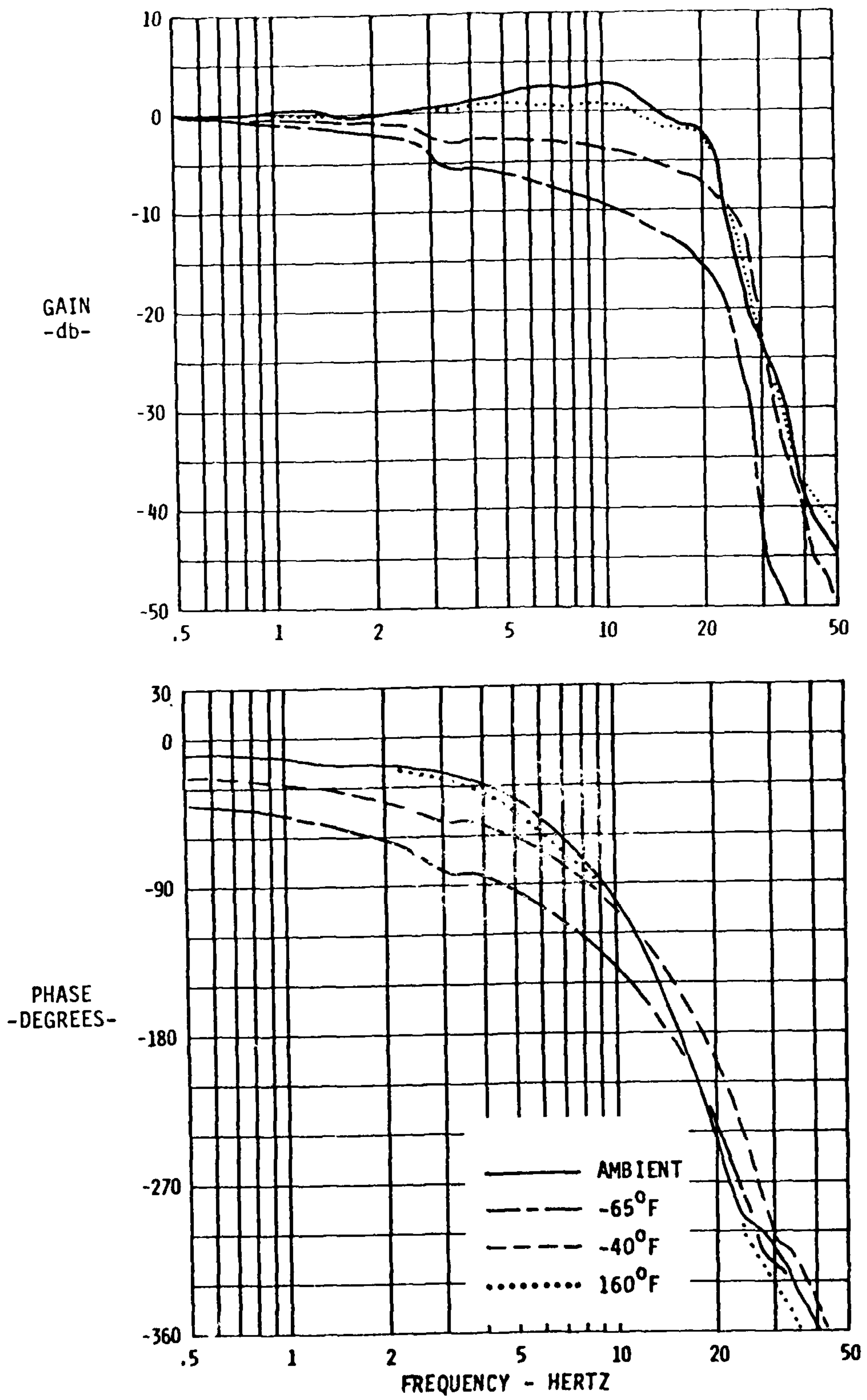


Figure E-12 Frequency Response, Standard Brake System, 66% ± 100 psi

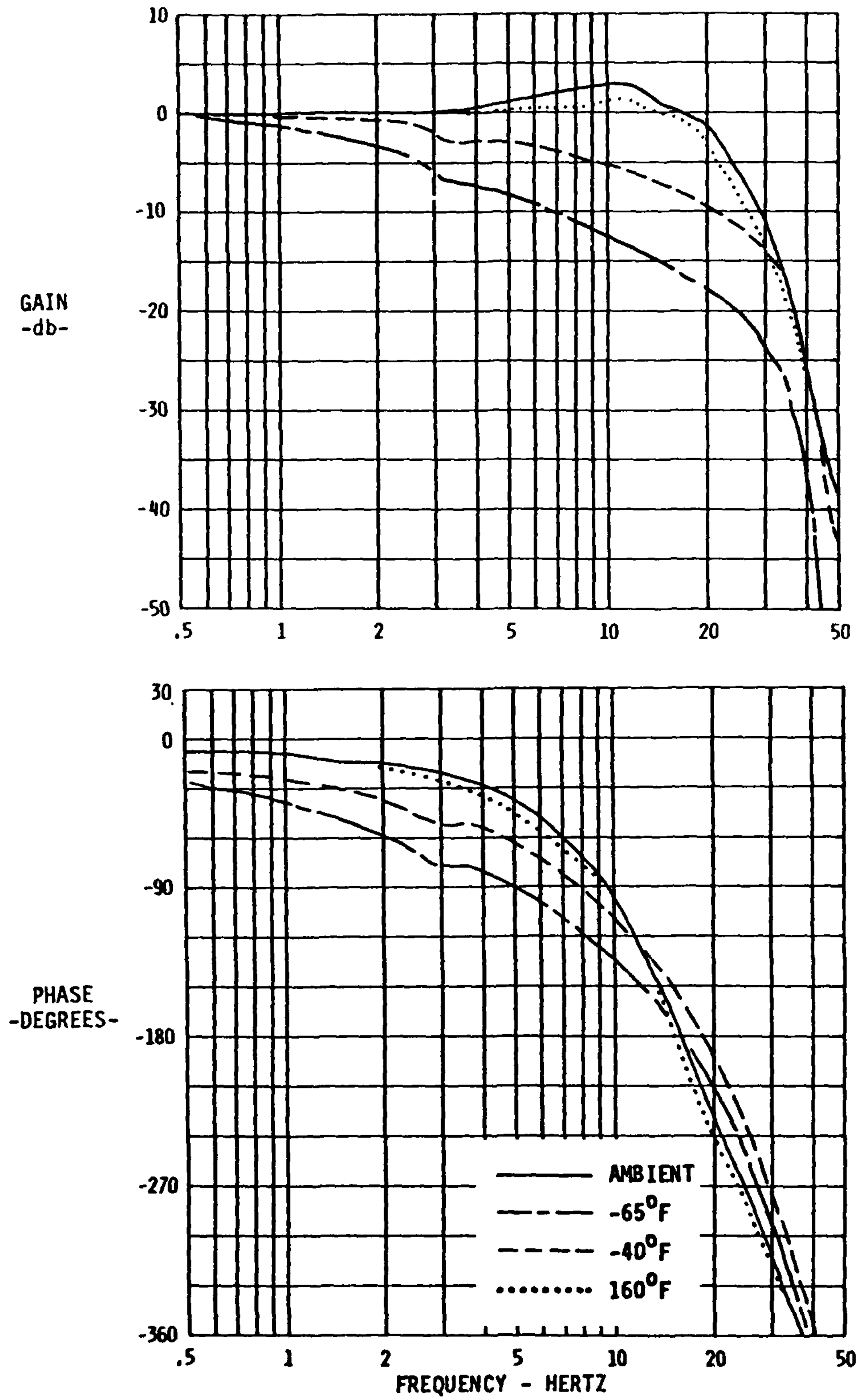
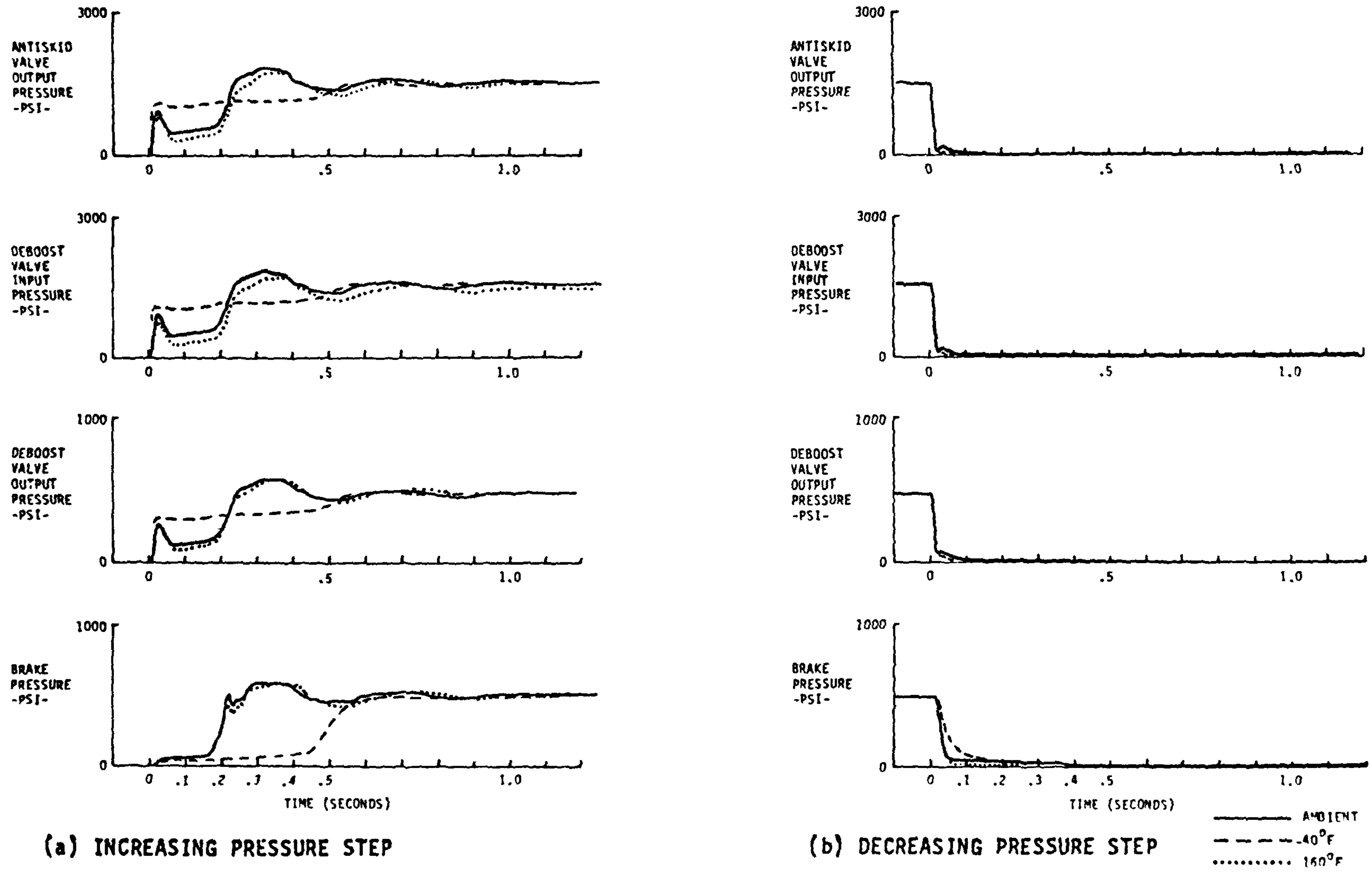


Figure E-13 Frequency Response, Standard Brake System, 66% ± 200 psi

TABLE E-4 STEP RESPONSE TEST CONDITIONS

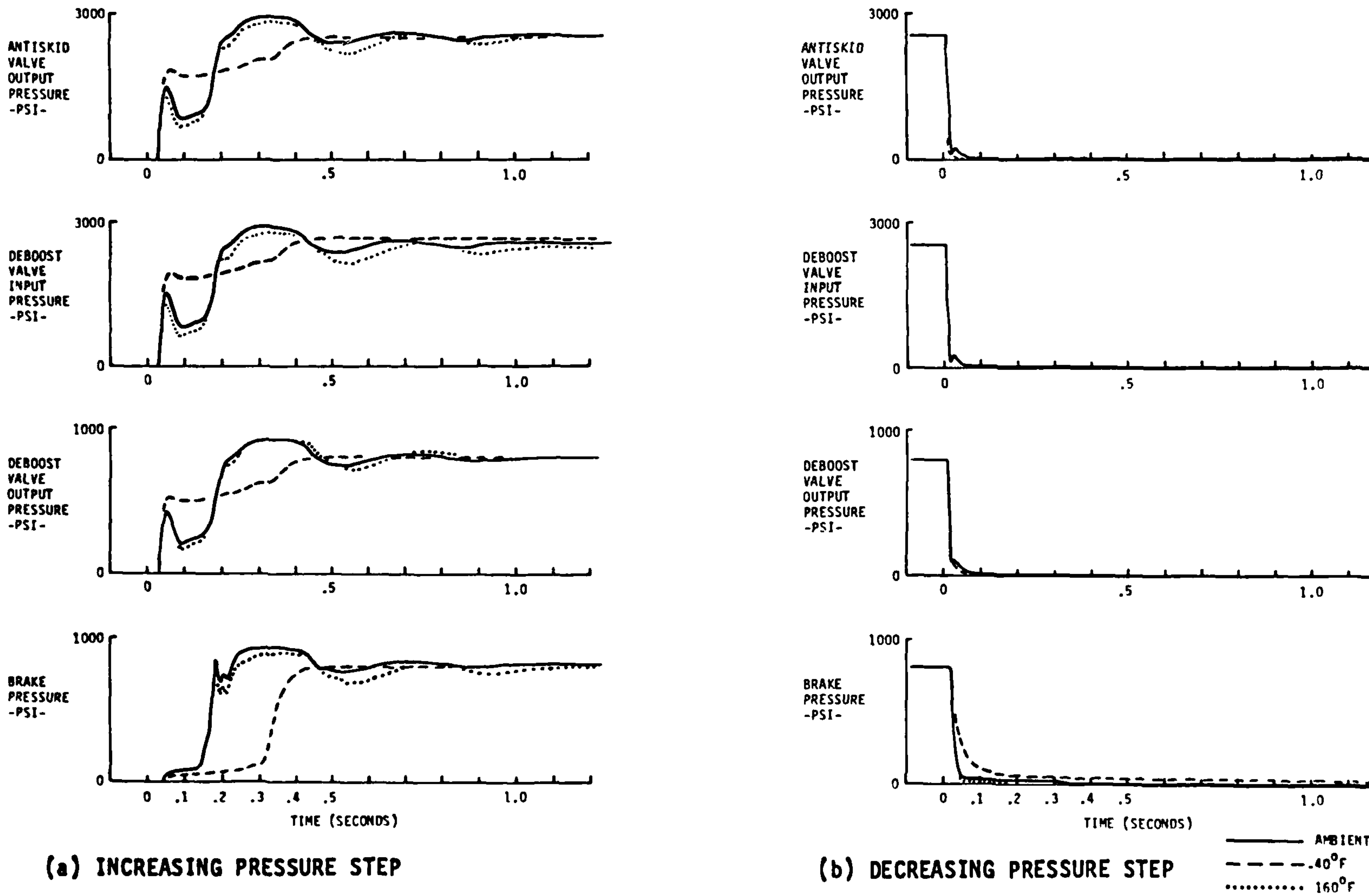
TEST	TEST TEMPERATURE	TEST POINTS INPUT	TEST POINTS OUTPUT	TEST CONDITION-% OF FULL BRAKE PRESSURE
2a. Increasing Pressure Step	Ambient	A	B,C,D,F	0 - 50
	-65°F			0 - 80
	160°F			0 - 100
				20 - 50
				20 - 80
				20 - 100
				50 - 80
				50 - 100
2b. Decreasing Pressure Step	Ambient	A	B,C,D,F	50 - 0
	-65°F			80 - 0
	160°F			100 - 0
				50 - 20
				80 - 20
				100 - 20
				80 - 50
				100 - 50



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

Figure E-14 Step Response, Standard System, 0-50%



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

Figure E-15 Step Response, Standard System, 0-80%

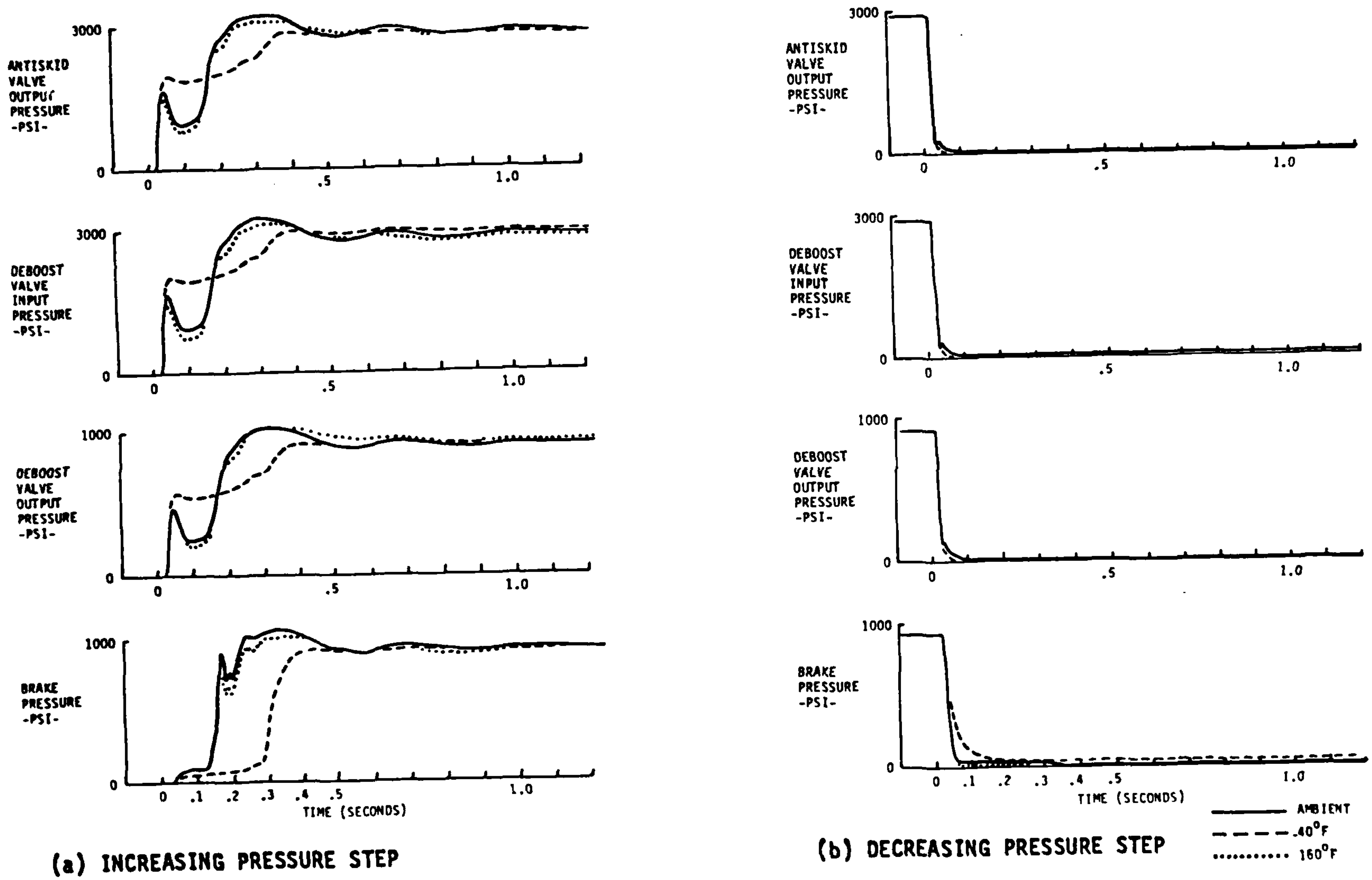


Figure E-16 Step Response, Standard System, 0-100%

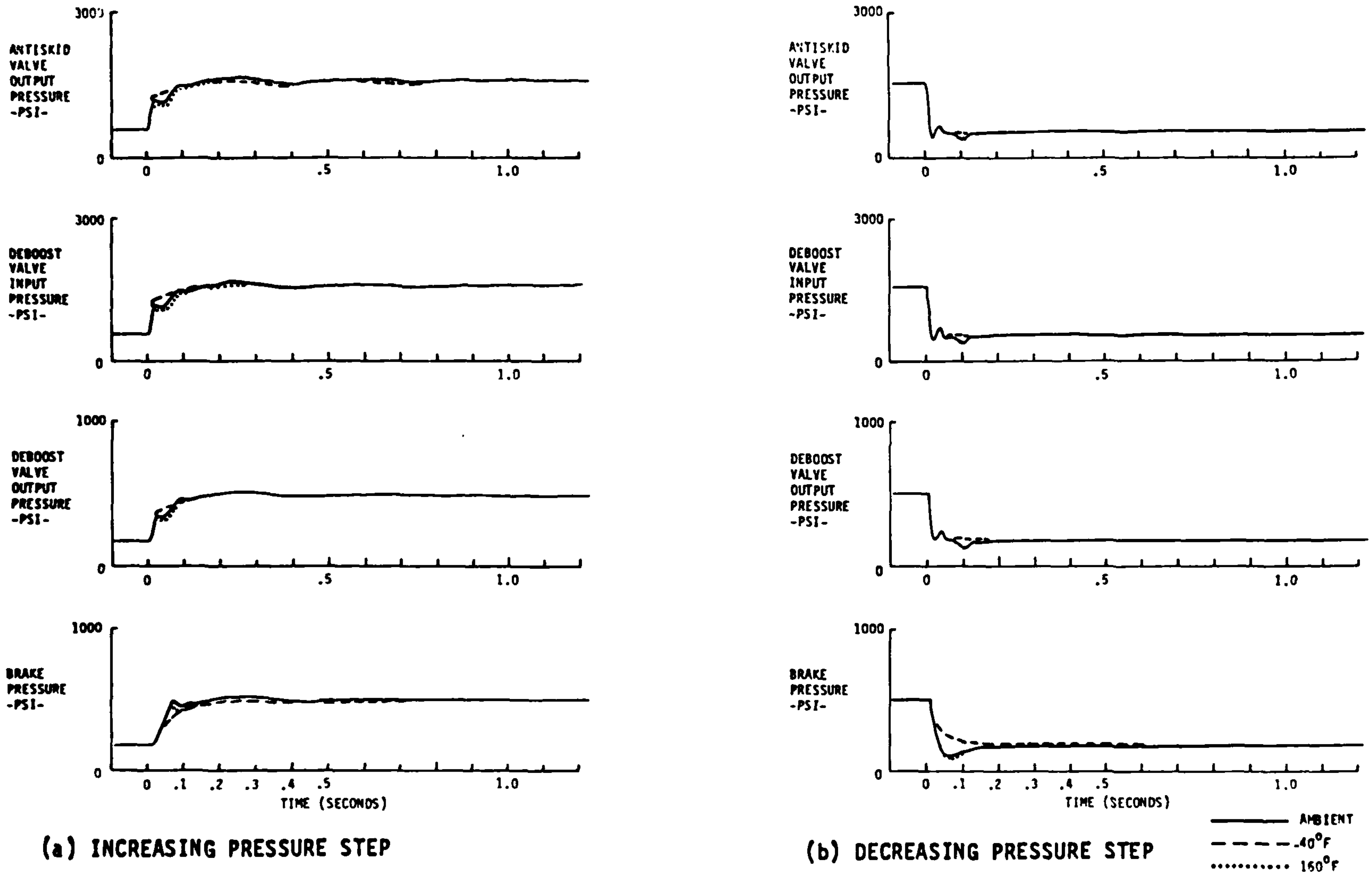
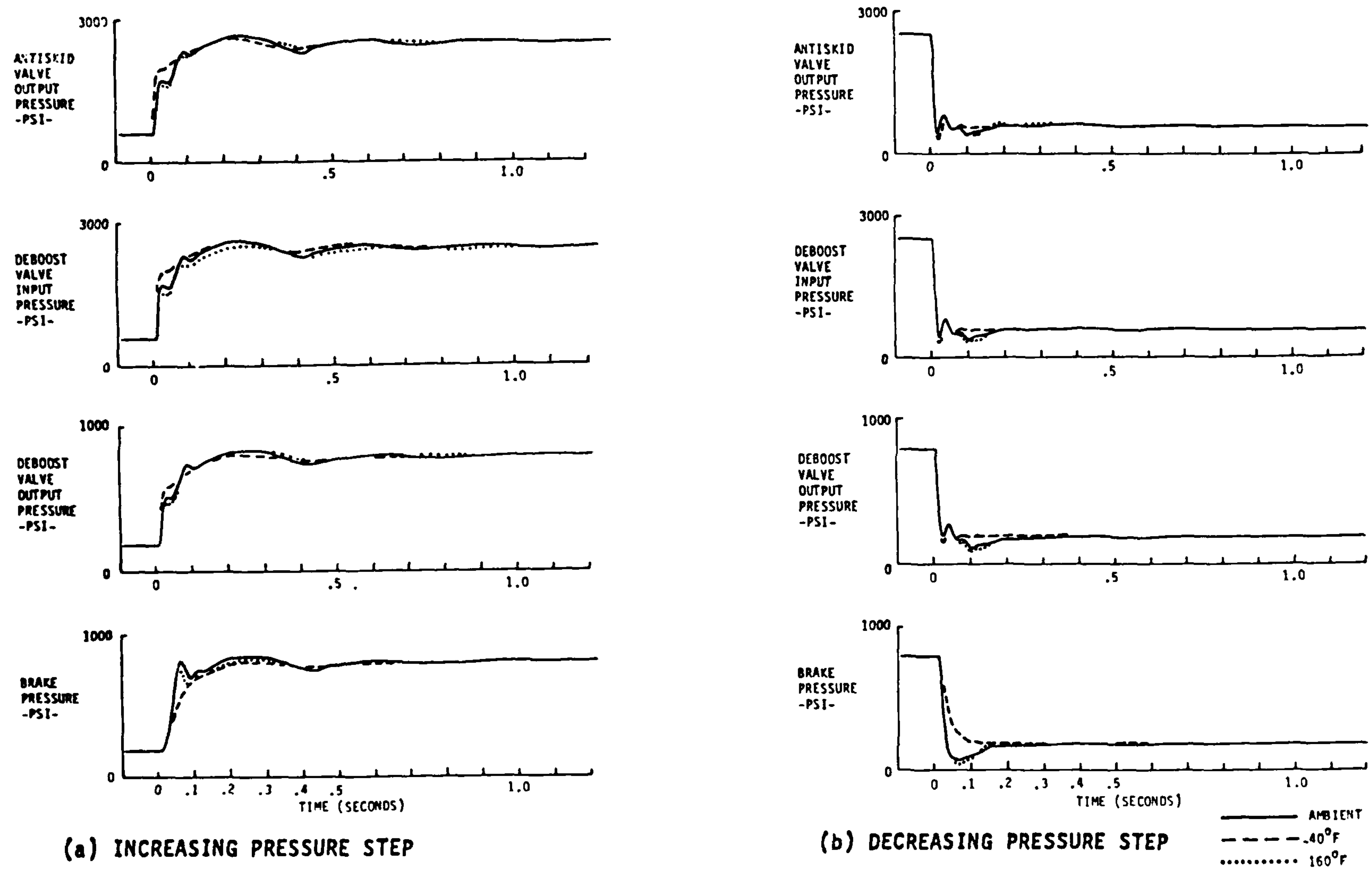


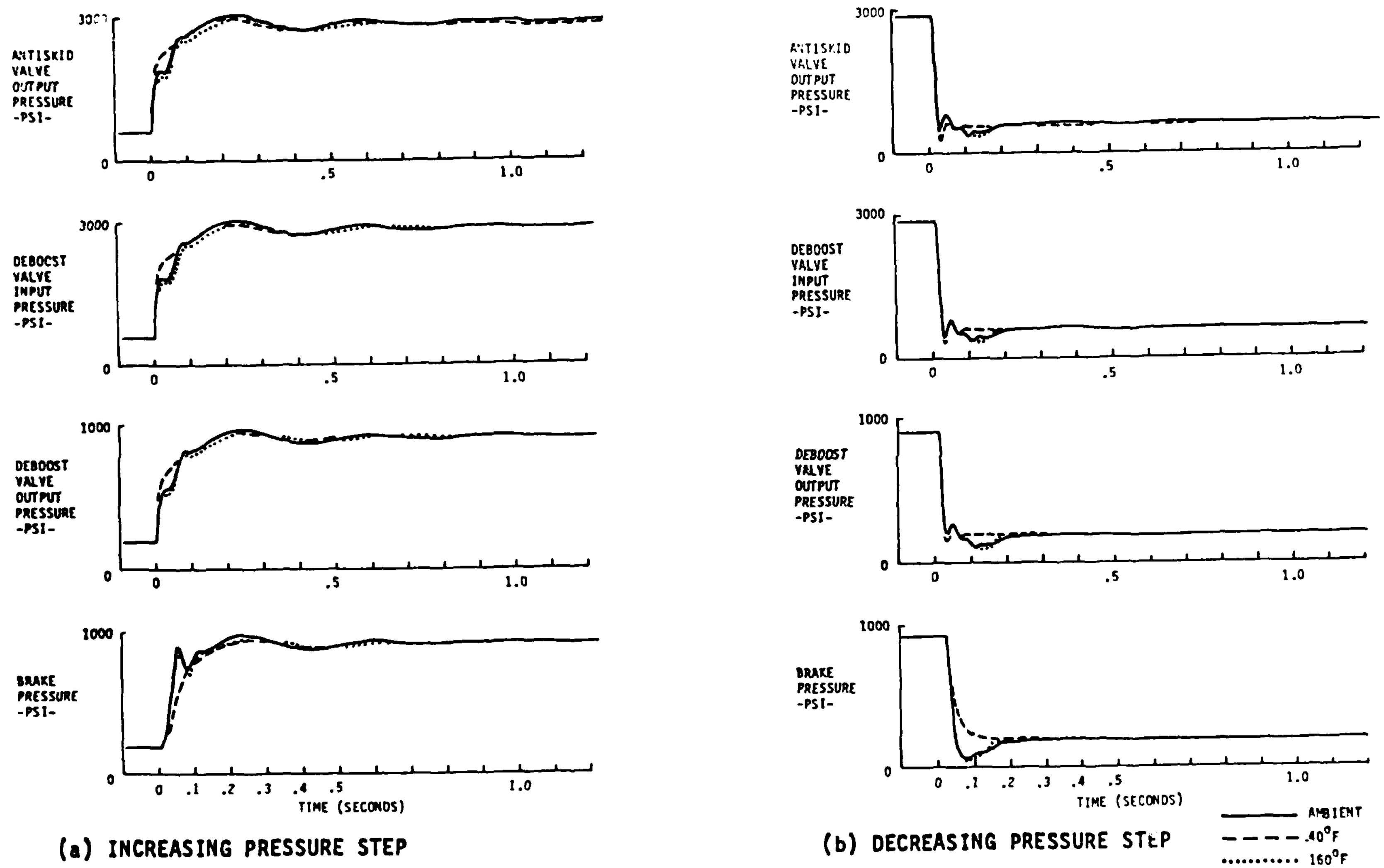
Figure E-17 Step Response, Standard System, 20-50%



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

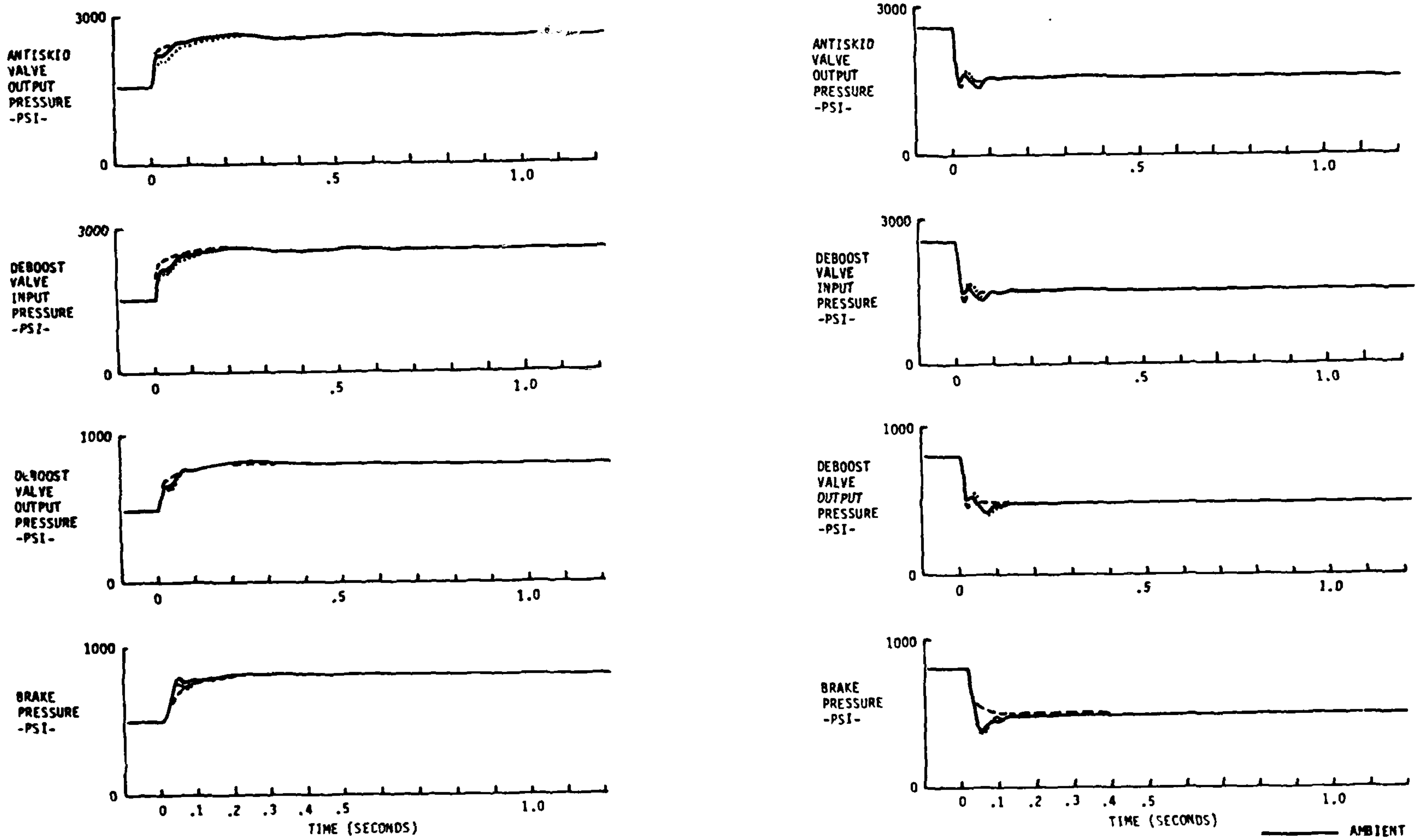
Figure E-18 Step Response, Standard System, 20-80%



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

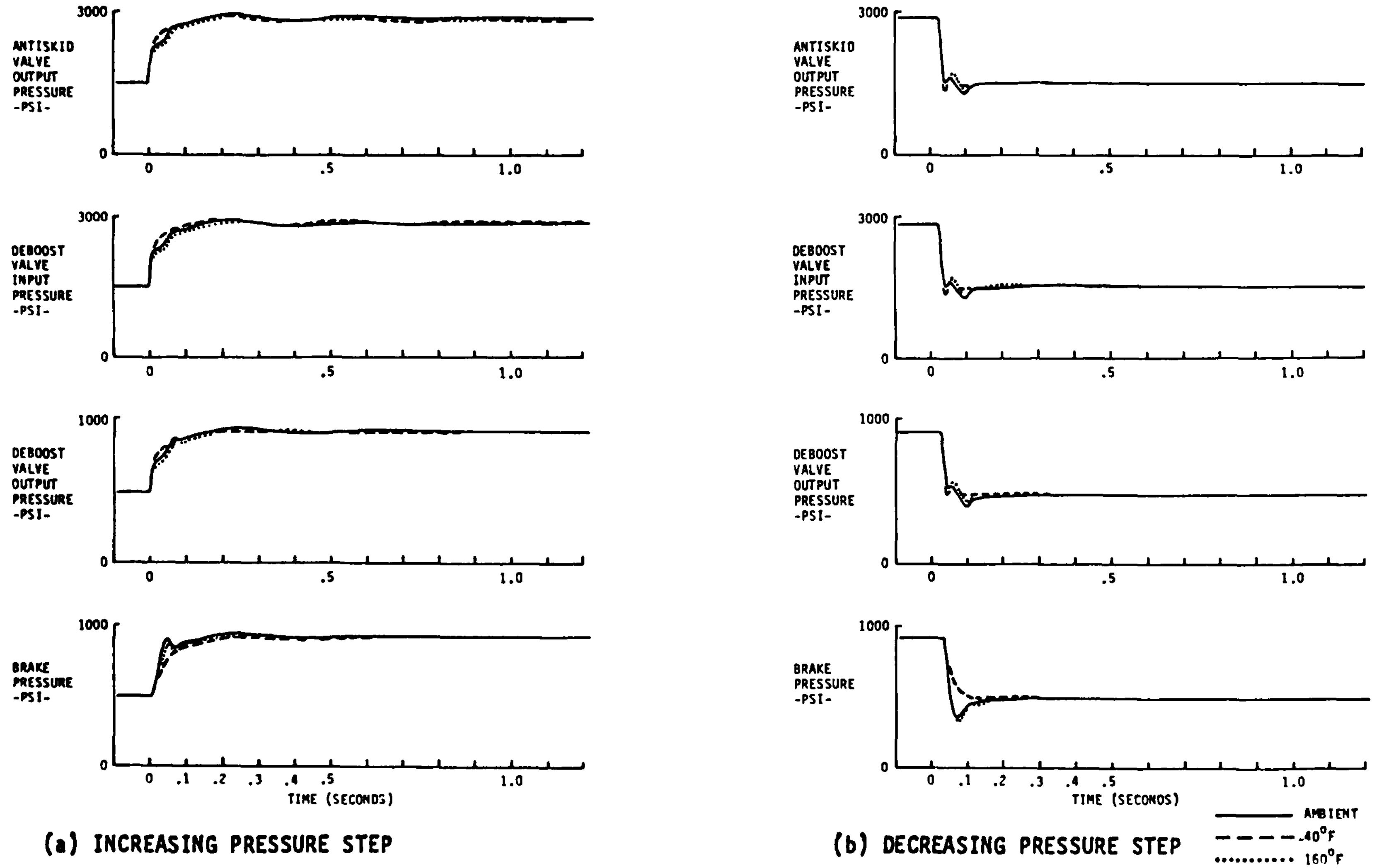
Figure E-19 Step Response, Standard System, 20-100%



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

Figure E-20 Step Response, Standard System, 50-80%



(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

Figure E-21 Step Response, Standard System, 50-100%

A 0.02 Hertz sinusoidal electrical control signal with a current amplitude of 0 to 50 milliamps was applied to the antiskid valve.

Brake pressure (test point E, Figure E-1) was recorded as a function of valve current. The pressure current characteristic was determined at three metered pressure levels (the pressure supplied to the antiskid valve) 33, 66 and 100 percent of full pressure. The test was performed at ambient temperature only. The test results are given in Figure E-22.

E.1.4 STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME, TEST 4

Brake pressure as a function of the fluid volume contained in the brake was measured to define the characteristics of the brake. The test was performed for reference only.

The brake was pressurized to approximately 965 psi. The pressure supply port of the brake was then closed. A small quantity of fluid was then bled from the brake into a graduated cylinder. The fluid volume and pressure were recorded. This bleed and recording procedure was repeated until the brake pressure was completely relieved. The test was performed at ambient temperature only. The test results are given in Figure E-23.

E.1.5 CONSTANT FRICTION RUNWAY, TEST 5

The stopping performance of the KC-135 aircraft was determined as a function of the runway friction coefficient.

During these tests braking was initiated two seconds after touchdown and continued until the aircraft decelerated to a typical turnoff velocity (24 feet per second). The peak available ground friction coefficient was held constant throughout the entire run. The distance travelled from brake application to 24 feet per second was recorded.

The test was performed at ambient, -65, -40 and +160 degrees Fahrenheit. The test results are given in Table E-5. Typical time history plots of wheel

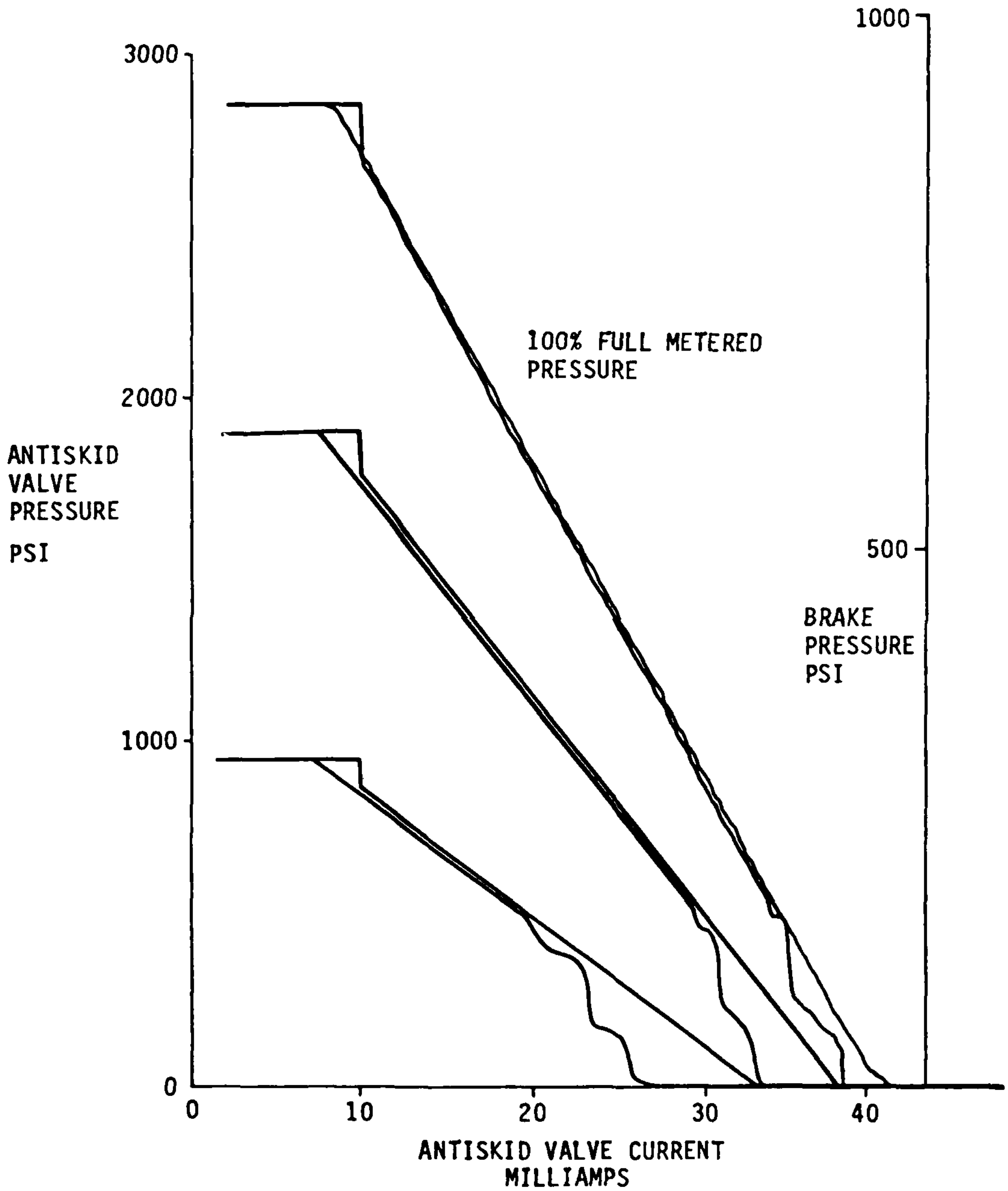


Figure E-22 Brake Pressure Versus Antiskid Valve Current

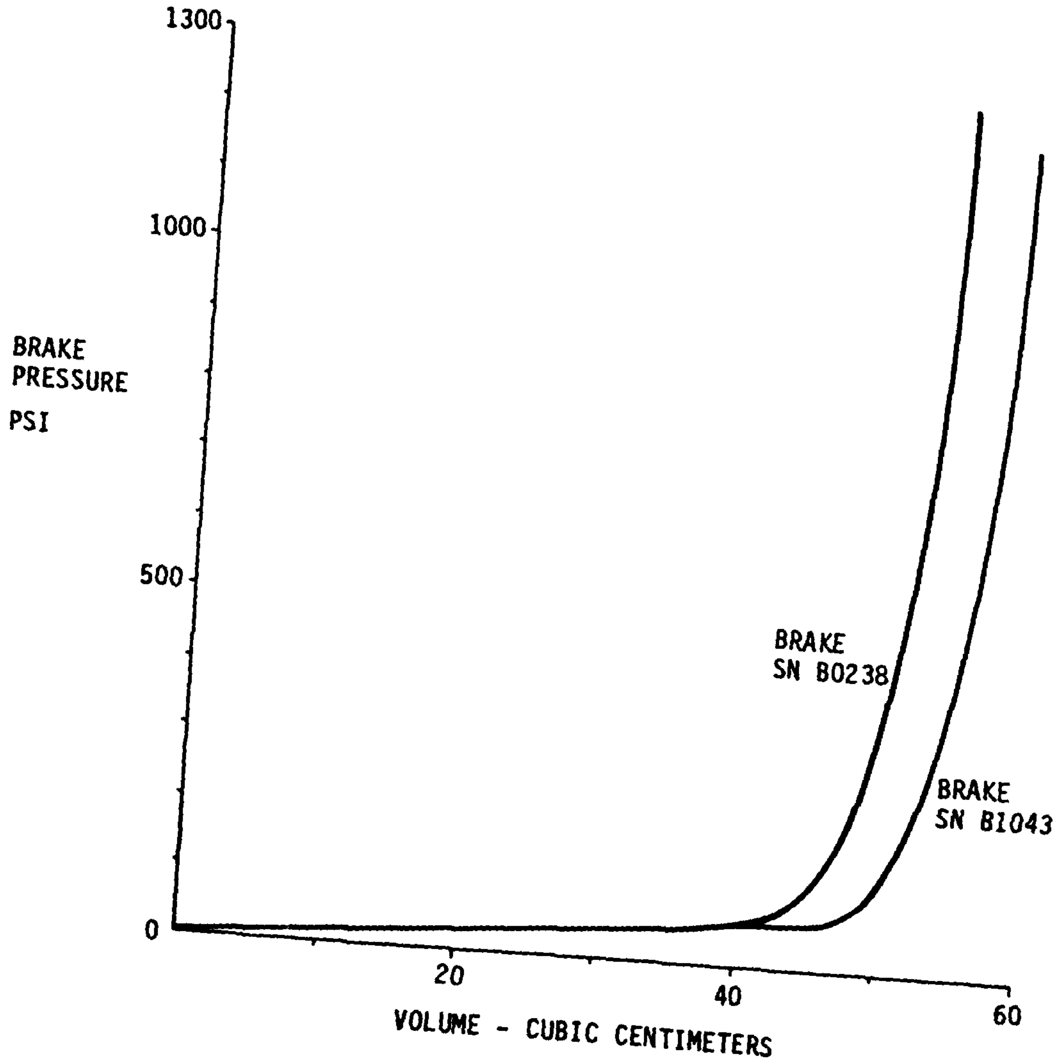


Figure E-23 Brake Pressure Versus Brake Volume, Standard Brakes

TABLE E-5 STOPPING DISTANCE, STANDARD SYSTEM

TEST/DESCRIPTION	FRICTION LEVEL	STOPPING DISTANCE -FEET (BRAKES ON TO 24 FPS) AMBIENT 73°F	-65°F	-40°F	160°F
TEST 5 CONSTANT RUNWAY FRICTION	.6	1879	2367	1895	1825
	.5	2234	2545	2240	2158
	.4	2694	3171	2779	2650
	.3	3784	4036	3548	3942
	.2	7214	5528	5452	6706
	.1	13325	9083	10396	12828
TEST 6 WET RUNWAY - FRICTION AS A FUNCTION OF AIRCRAFT VELOCITY					
	.1 - .5	4745	4486	4331	4608
	.1 - .35	5963	5391	5208	5936
TEST 7 STEP FRICTION					
	.1 - .5	3957	6732	6105	3307

speed, brake pressure and antiskid valve current at runway friction coefficients (MU) of .5, .3 and .1, and ambient temperature are given in Figures E-24, E-25 and E-26. Similiar results at low temperature (-40 degrees Fahrenheit) and high temperature (+160 degrees Fahrenheit) are given in Figures E-27 thru E-32.

E.1.6 WET RUNWAY, TEST 6

The stopping performance and adaptability of the KC-135 brake system to a slowly changing runway friction condition (simulating wet runway operation) was determined.

The peak available ground friction coefficient was varied (in a linear fashion as a function of aircraft velocity) from a low valve at high speed to a high valve at low speed, see Figure E-33. The test was performed for two levels of wet runway friction, .1 to .5 and .1 to .35. The distance travelled from brake application to 24 feet per second was recorded.

The test was performed at ambient, -65, -40 and +160 degrees Fahrenheit. The test results are given in Table E-5. Typical time history plots of wheel speed, brake pressure and antiskid valve current for the .1 to .5 friction case at ambient, -40 and +160 degrees Fahrenheit are given in Figures E-34, E-35 and E-36. Time history data for the .1 to .35 case are similar to those shown.

E.1.7 STEP FRICTION, TEST 7

The stopping performance and adaptability of the KC-135 brake system to a step change in runway friction (simulating icy patches or tar strips) was determined.

During a normal braked landing, the peak available runway friction coefficient was varied in the step fashion shown in Figure E-37. The distance from brake application to 24 feet per second was recorded.

The test was performed at ambient, -65, -40 and +160 degrees Fahrenheit. The

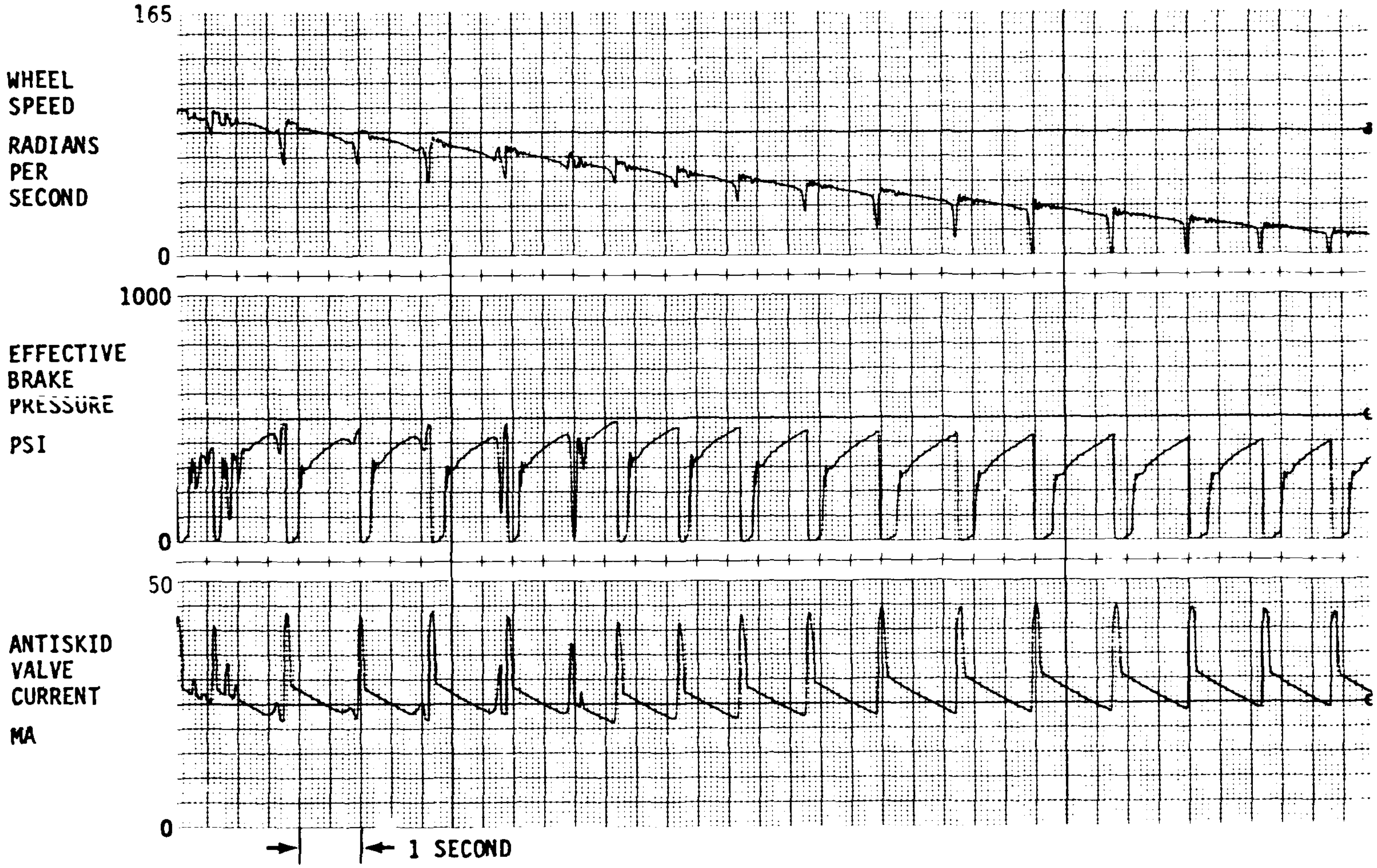


Figure E-24 Brake System Performance, Standard System, MU=.5, Ambient

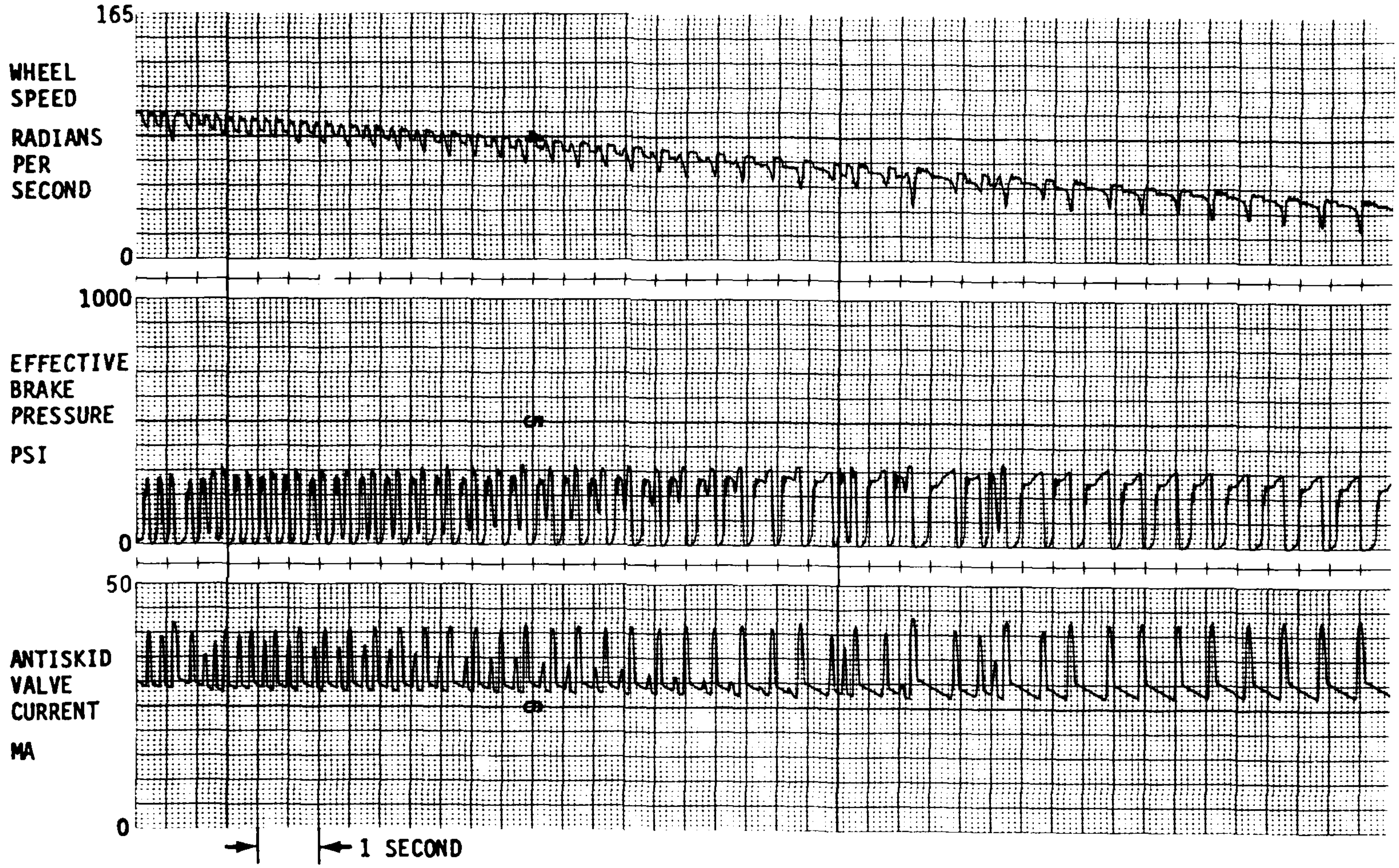


Figure E-25 Brake System Performance, Standard System, MU=.3, Ambient

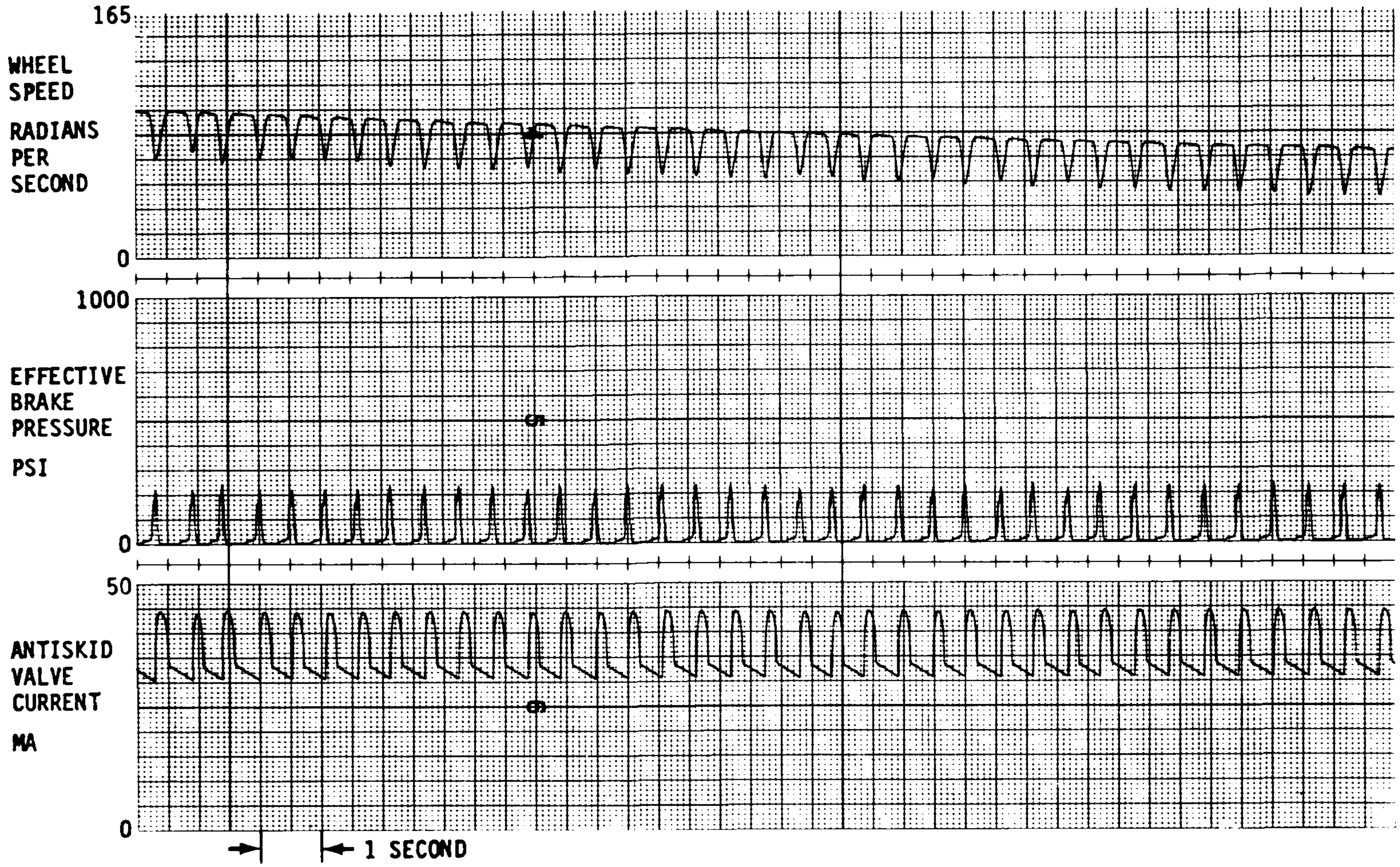


Figure E-26 Brake System Performance, Standard System, MU=.1, Ambient

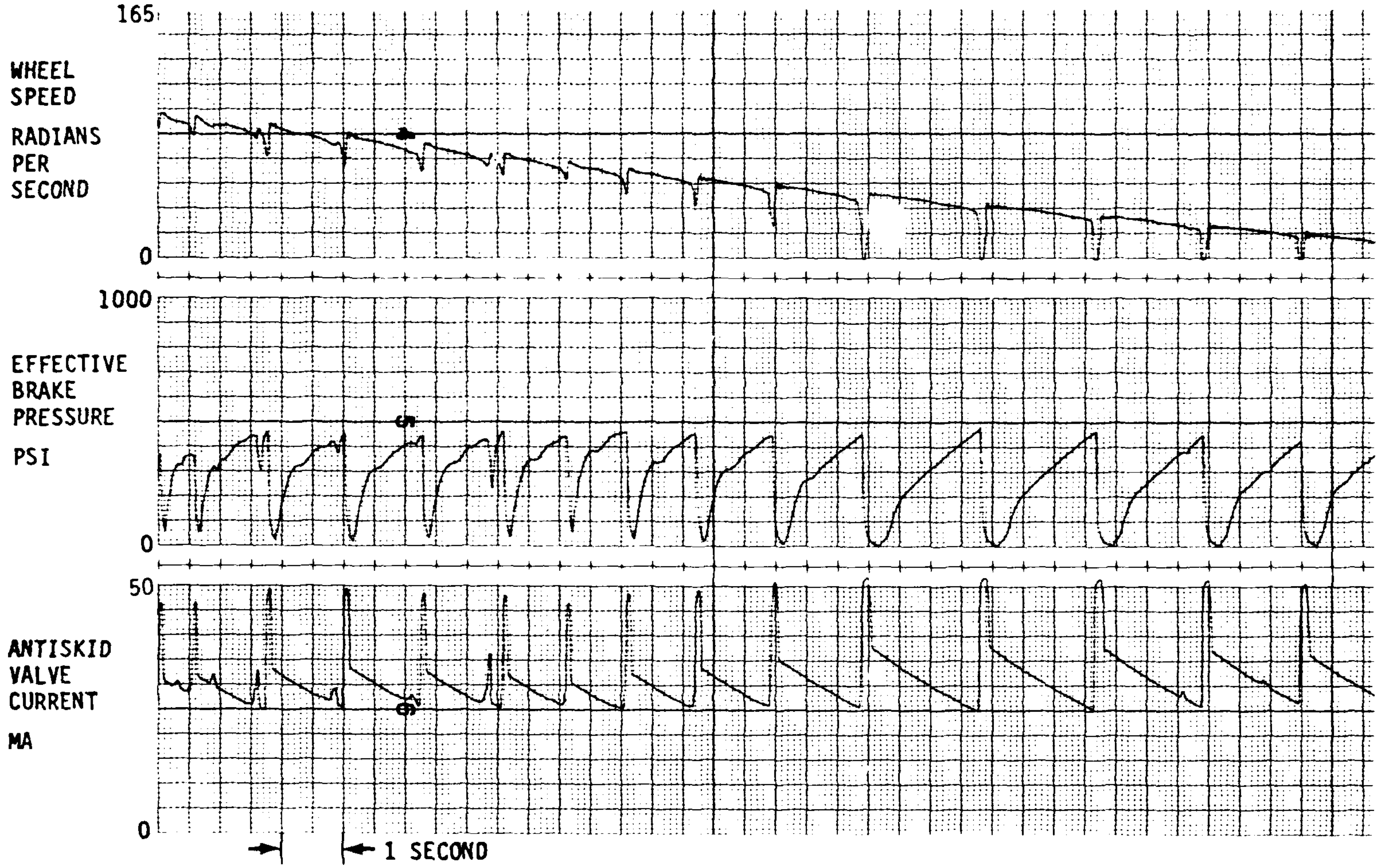


Figure E-27 Brake System Performance, Standard System, MU=.5, -40°F

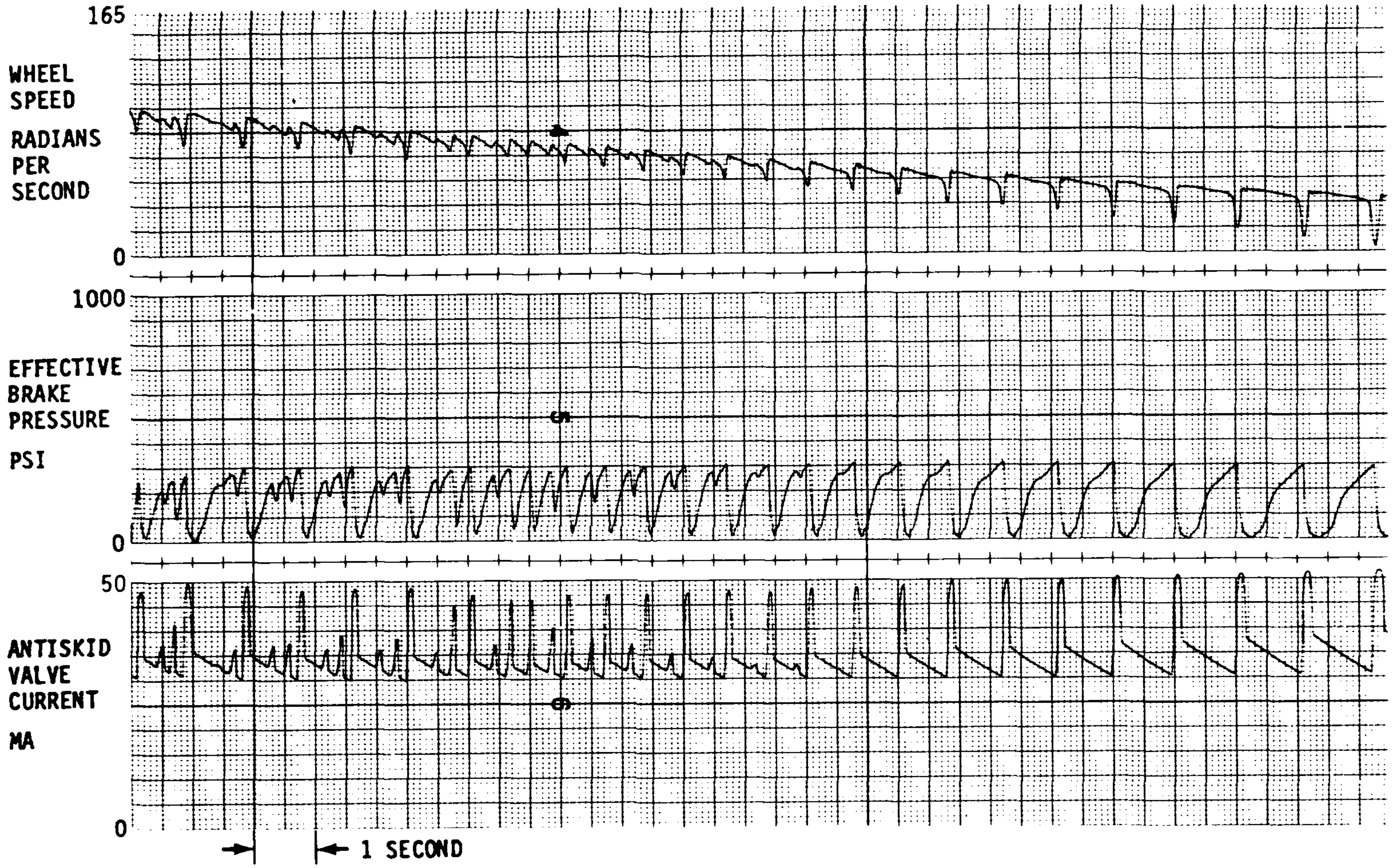


Figure E-28 Brake System Performance, Standard System, MU=.3, -40°F

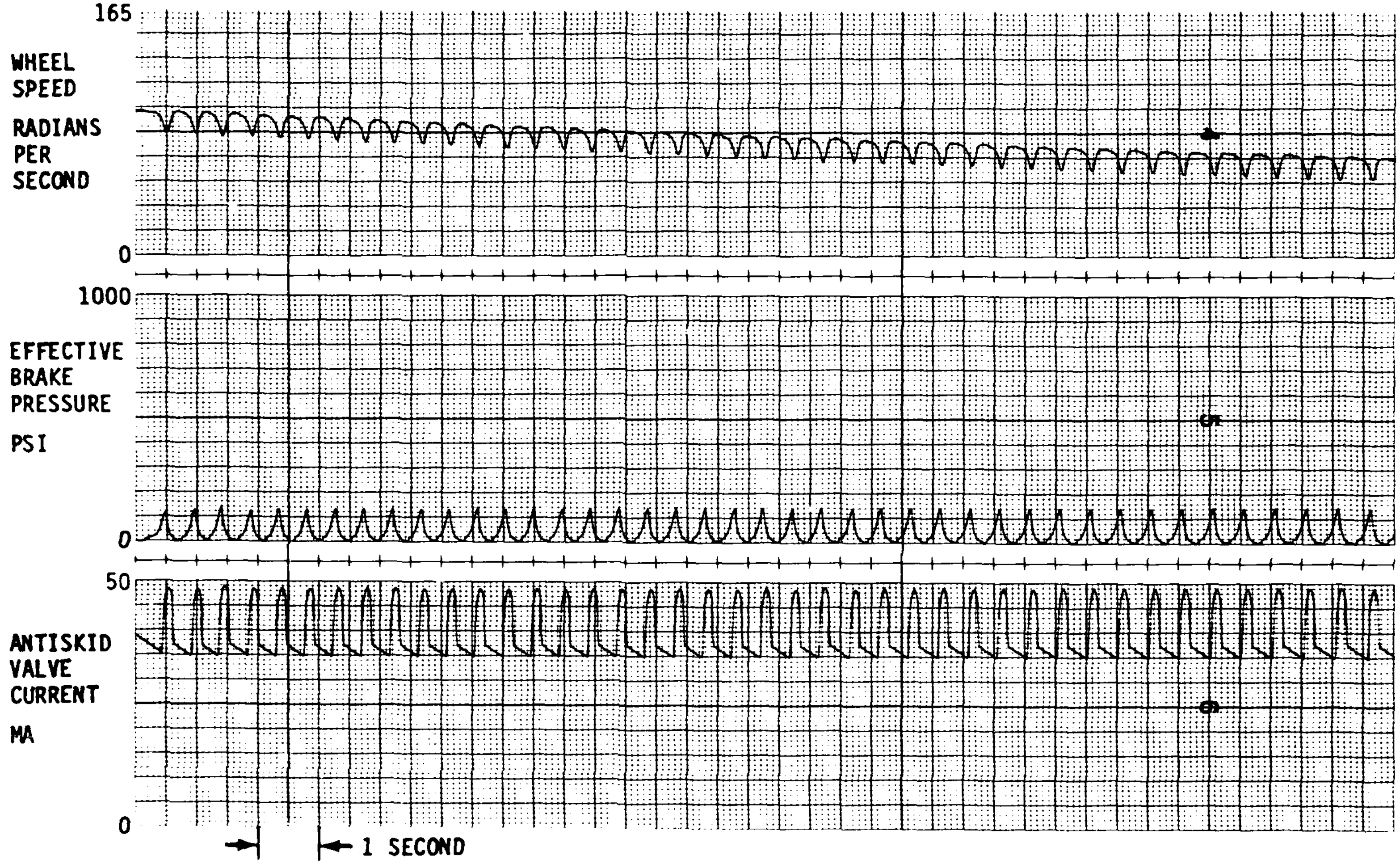


Figure E-29 Brake System Performance, Standard System, MU=.1, -40°F

Figure E-30 Brake System Performance, Standard System, MU=.5, 160°F

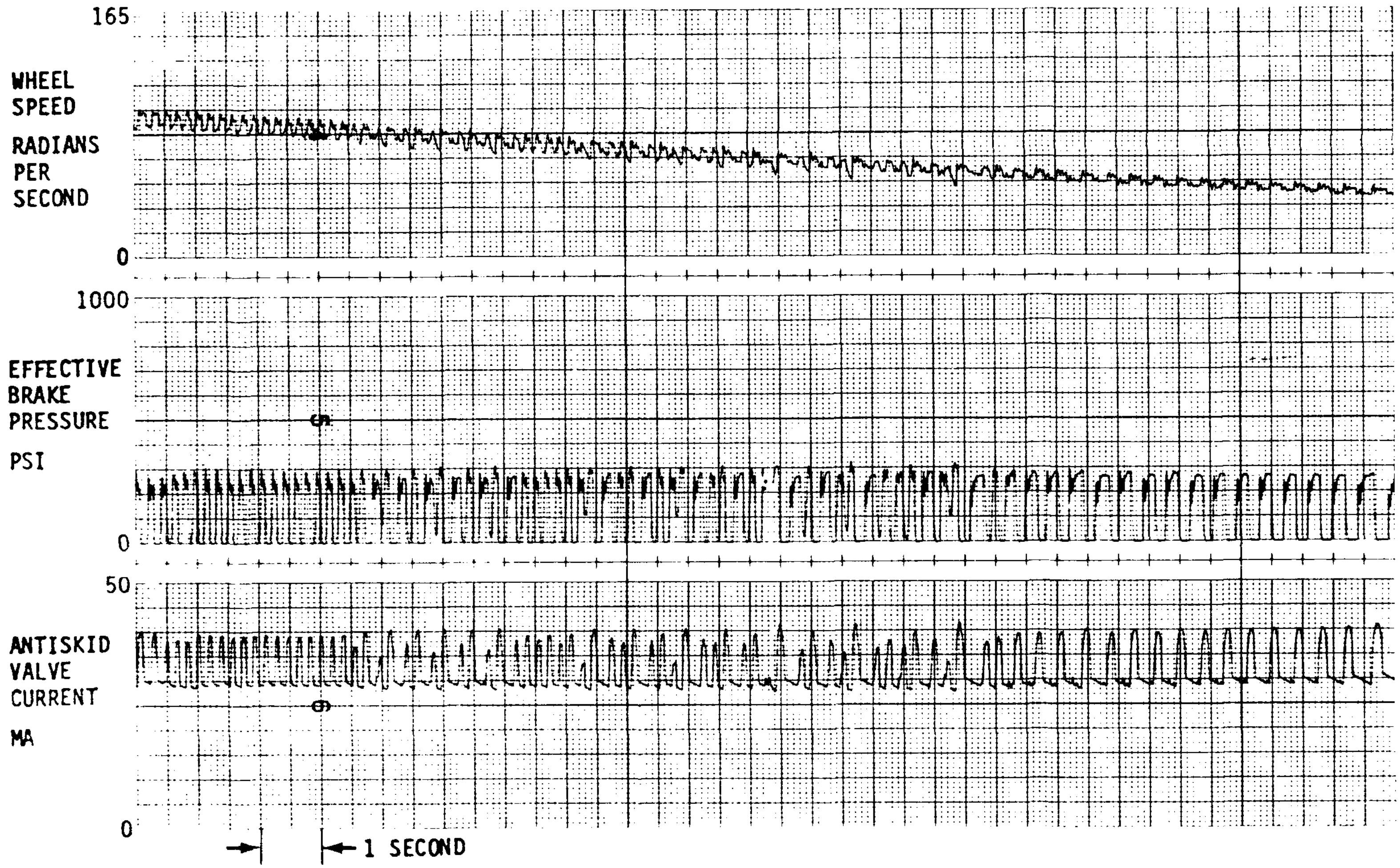


Figure E-31 Brake System Performance, Standard System, MU=.3, 160°F

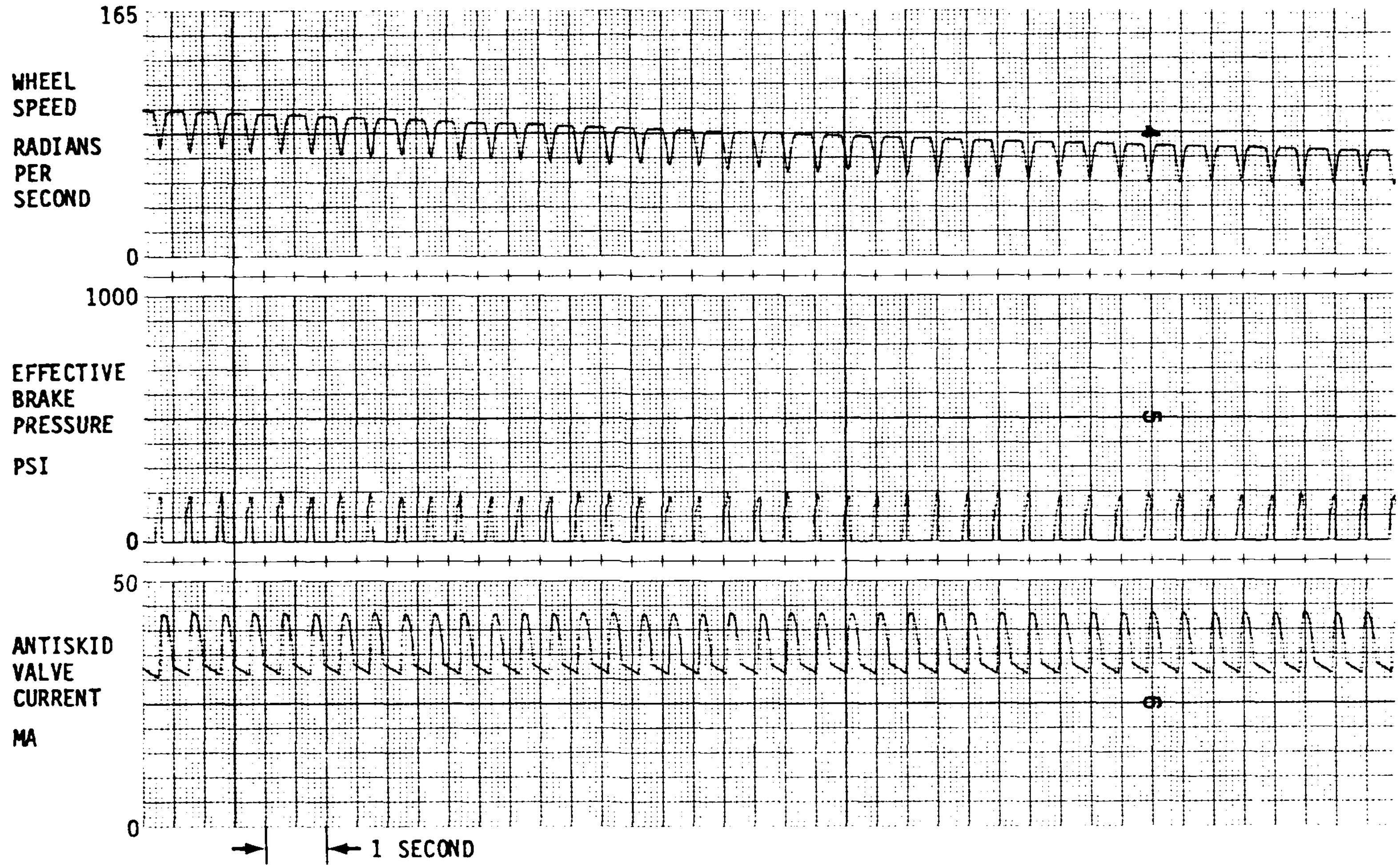
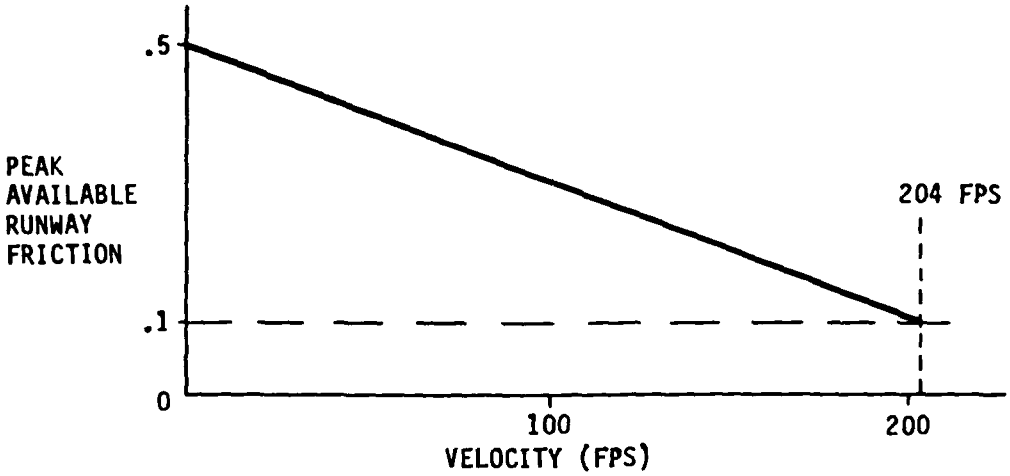
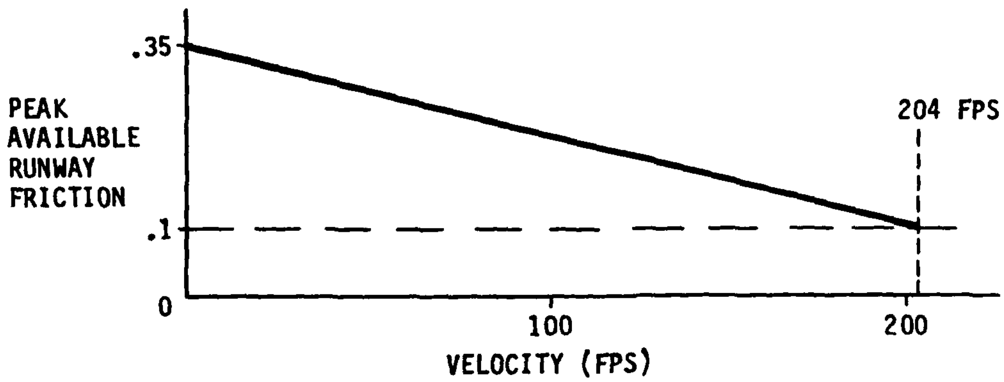


Figure E-32 Brake System Performance, Standard System, MU=.1, 160°F



a) CONDITION 1 - FRICTION LEVEL .1 TO .5



b) CONDITION 2 - FRICTION LEVEL .1 TO .35

Figure E-33 Wet Runway Test Conditions

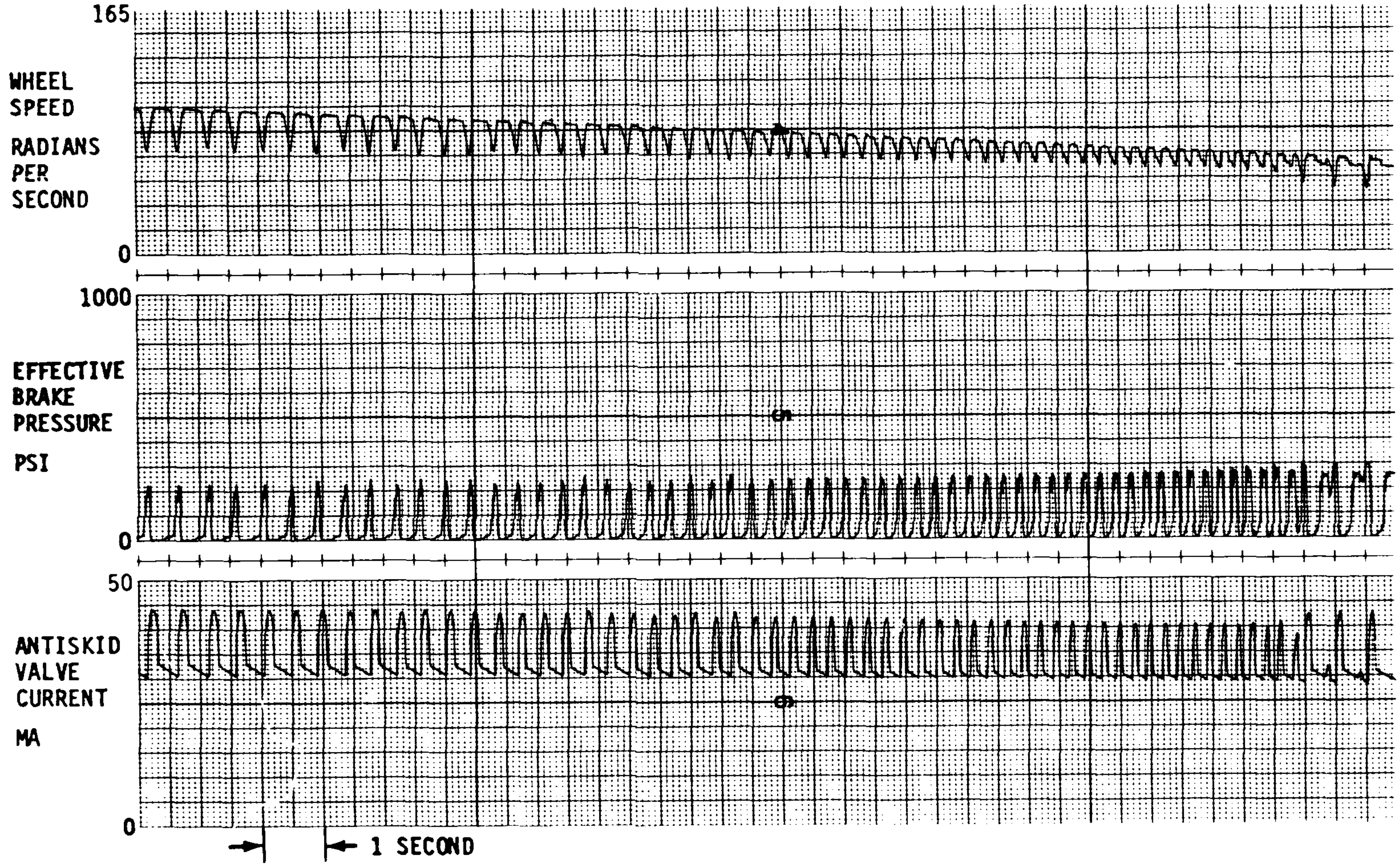


Figure E-34 Wet Runway Performance, Standard System, MU=.1 to .5, Ambient

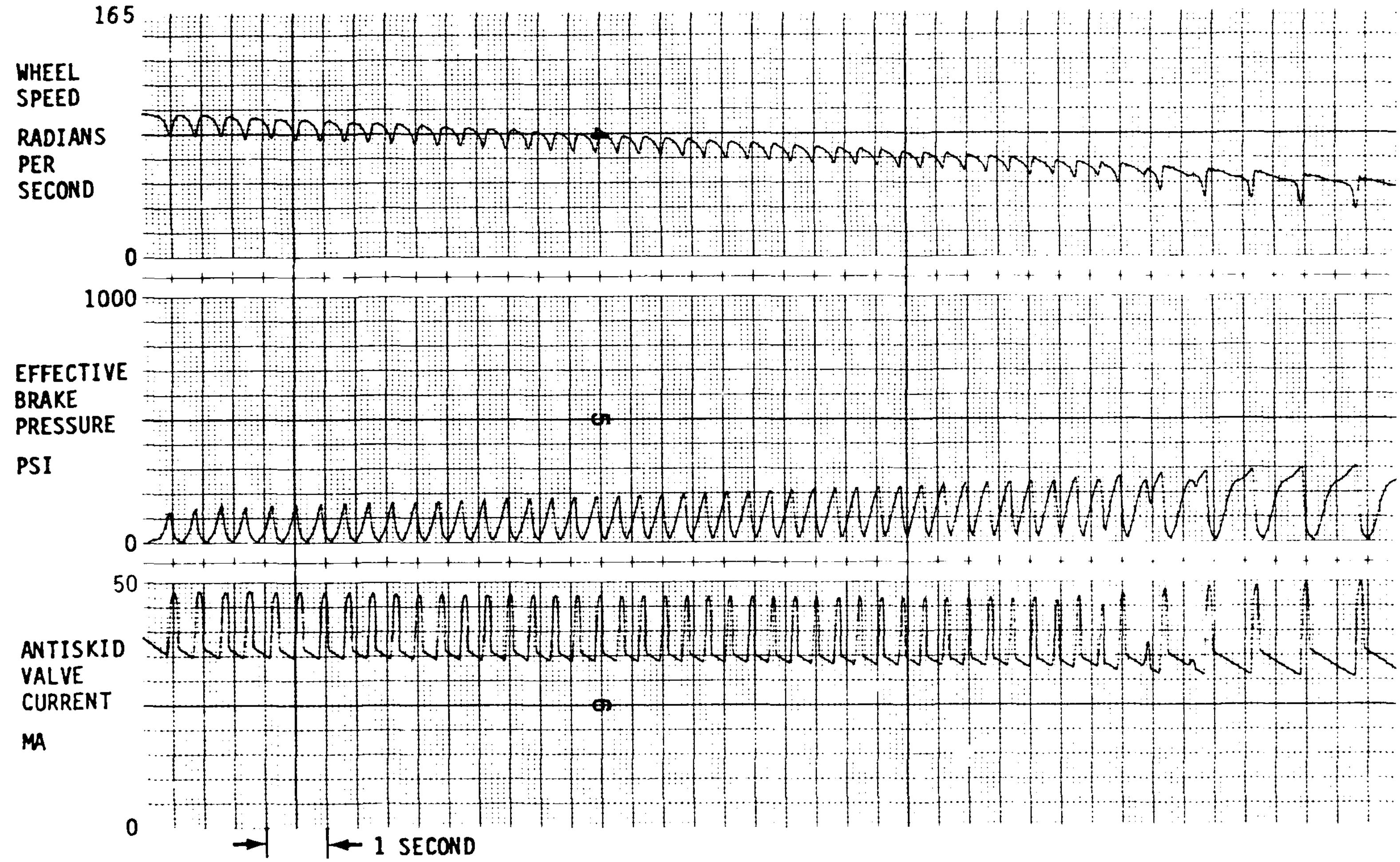


Figure E-35 Wet Runway Performance, Standard System, MU=.1 to .5, -40°F

AD-A111 319 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 1/2
FIREPROOF BRAKE HYDRAULIC SYSTEM.(U)
SEP 81 S M WARREN, J R KILNER F33615-80-C-2026
UNCLASSIFIED AFWAL-TR-81-2080 NL

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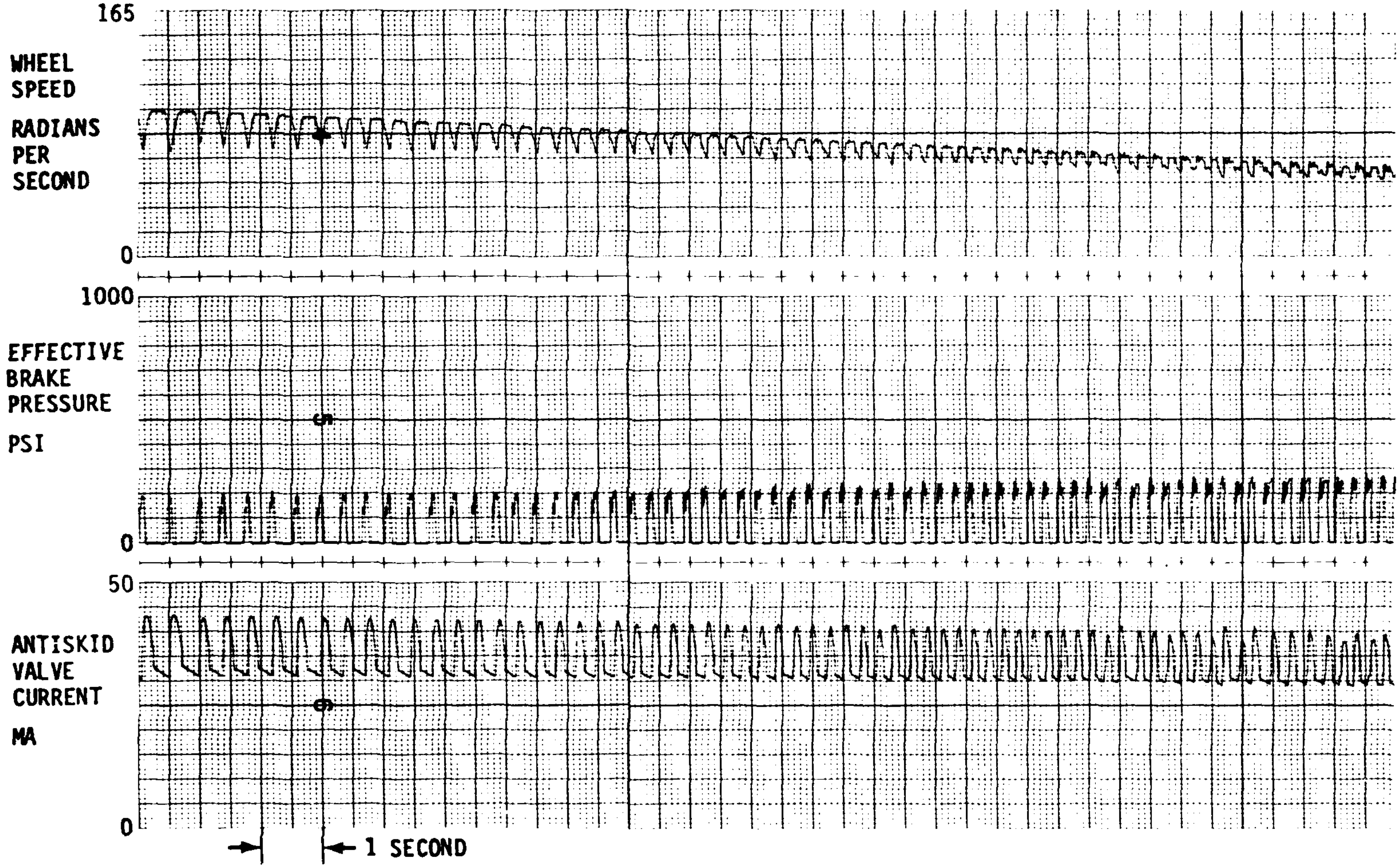


Figure E-36 Wet Runway Performance, Standard System, MU=.1 to .5, 160°F

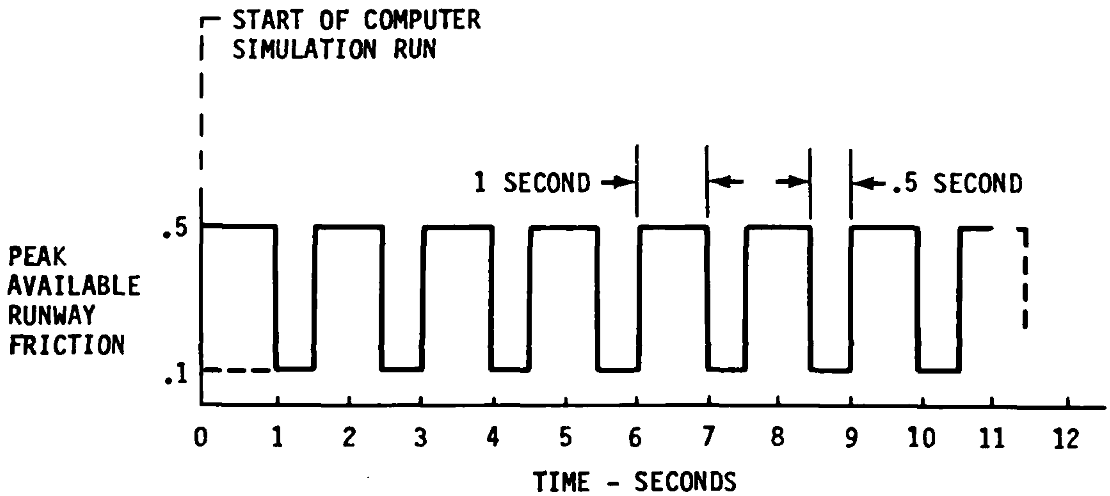


Figure E-37 Step Friction Test Condition

stopping distance test results are given in Table E-5. Typical time history plots of wheelspeed, brake pressure, antiskid valve current and the peak runway friction coefficient at ambient, -40 and +160 degrees Fahrenheit are given in Figures E-38, E-39 and E-40.

E.1.8 LANDING GEAR STABILITY, TEST 8

The extent to which the KC-135 brake control system contributes to the fore and aft vibrational stability of the landing gear was evaluated by determining the minimum level of fore and aft landing gear strut damping required for stable landing gear oscillations.

During a normal braked landing (at a runway friction coefficient of 0.5) the landing gear strut was made to oscillate by increasing the brake torque to 1.5 times its normal value for a short period of time. The strut damping ratio was lowered until the landing gear oscillations were undamped, the brake system unstable or the strut damping ratio was zero. The strut damping ratio at the point of instability was recorded.

The test was performed at ambient, -65, -40 and +160 degrees Fahrenheit. The test results are given in Table E-6. Typical time history plots of wheel speed, brake pressure, valve current, ground force, brake torque and strut displacement at ambient temperature with normal strut damping (damping ratio equals .1) and zero damping are given in Figures E-41 and E-42. The strut is stable in both of these cases. At low temperature and with a damping ratio of zero, the strut oscillations are undamped. Time history data at -40 degrees Fahrenheit and zero strut damping are shown Figure E-43.

E.2 TWO-FLUID BRAKE HYDRAULIC SYSTEM PERFORMANCE TEST RESULTS

Performance testing of the two-fluid brake hydraulic syste[illegible] performed with the hydraulic system mockup shown in Figures 17 and E-44. The [illegible]kup employs the KC-135 brakes and deboost valve which were modified for use in the two-fluid brake system. The configuration of the active portion of the mockup is in identical to the standard KC-135 brake hydraulic system mockup (Table E-1). Additional hydraulic tubing, fittings and an air operated hydraulic

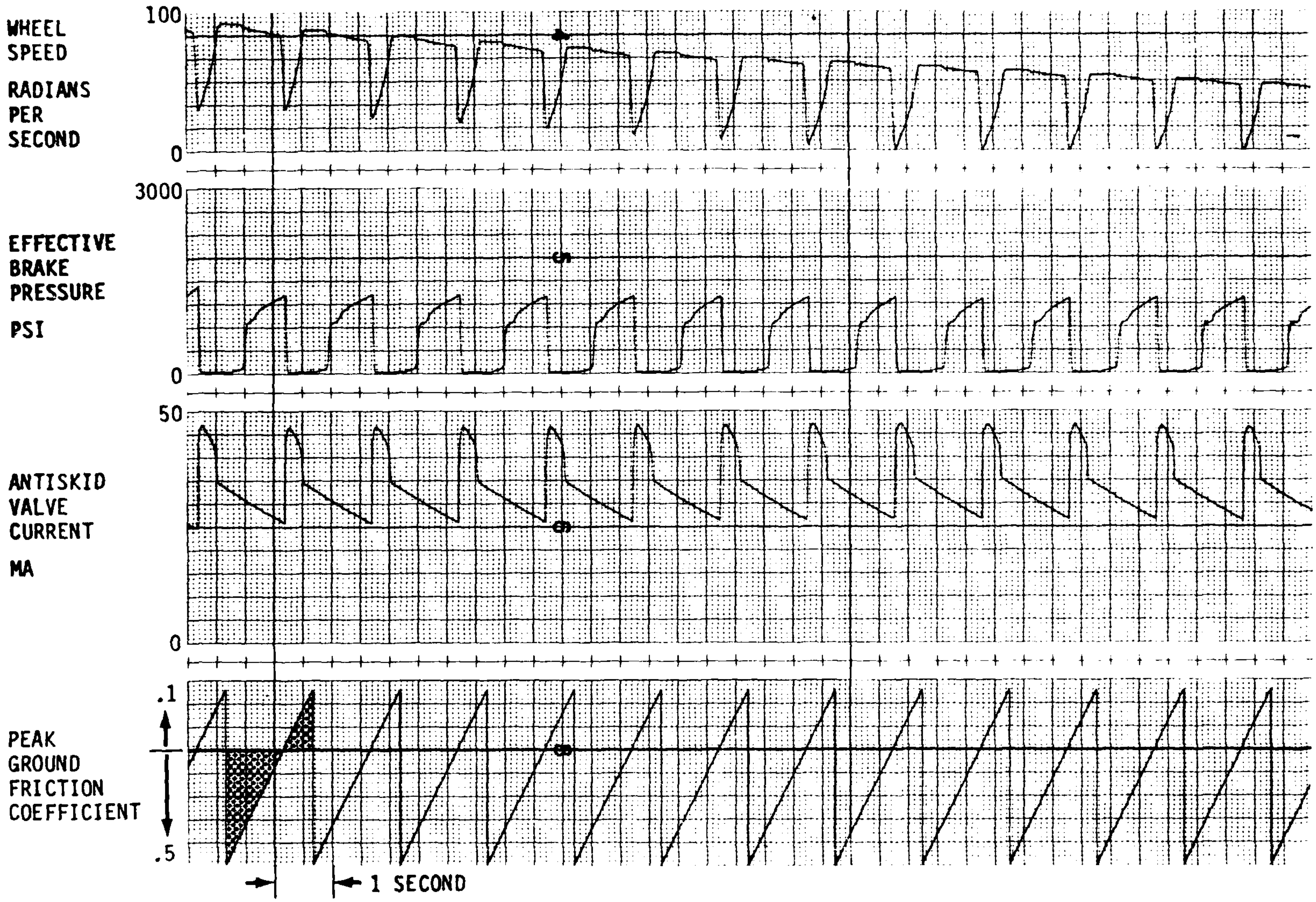


Figure E-38 Step Friction Performance, Standard System, Ambient

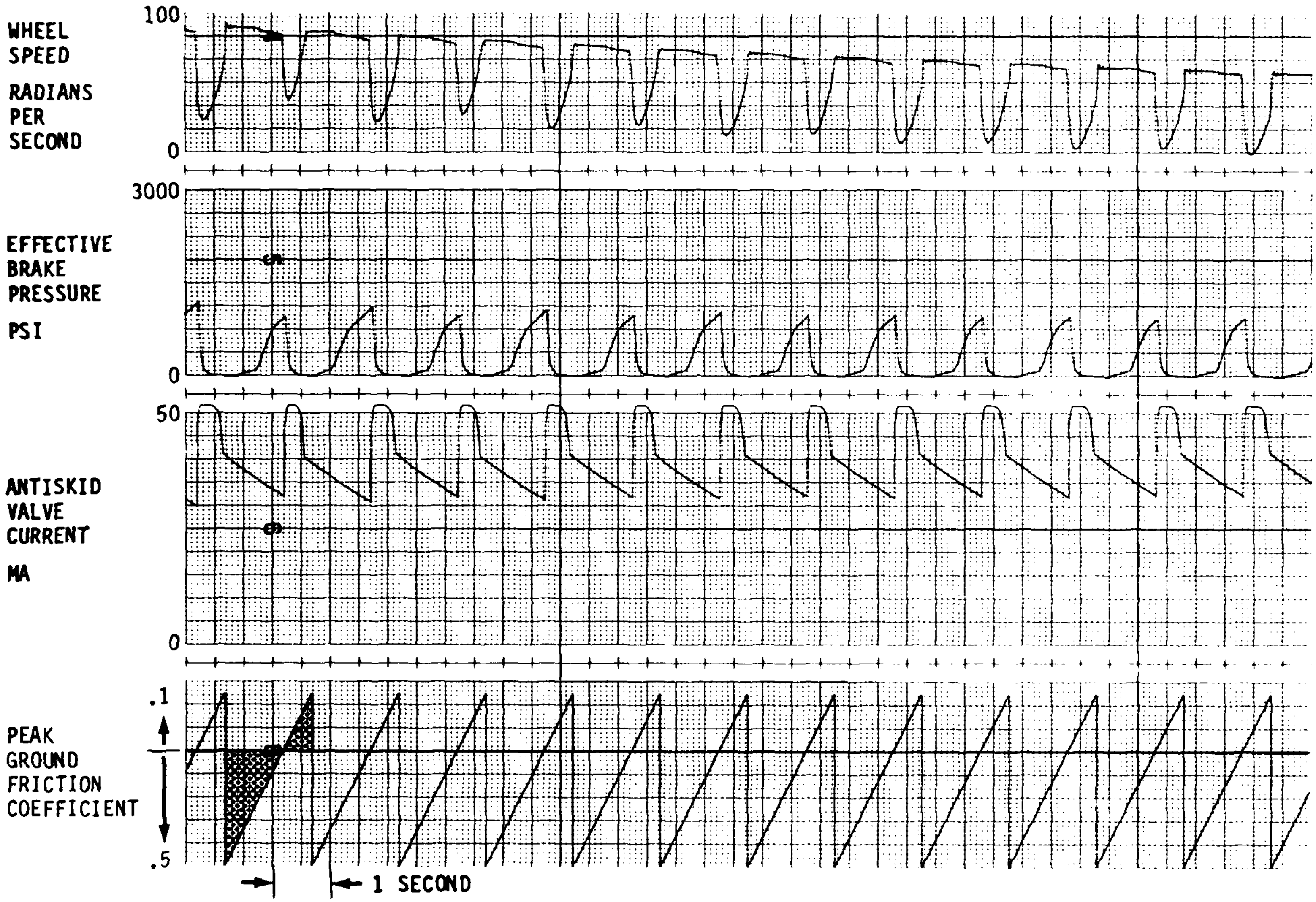


Figure E-39 Step Friction Performance, Standard System, -40°F

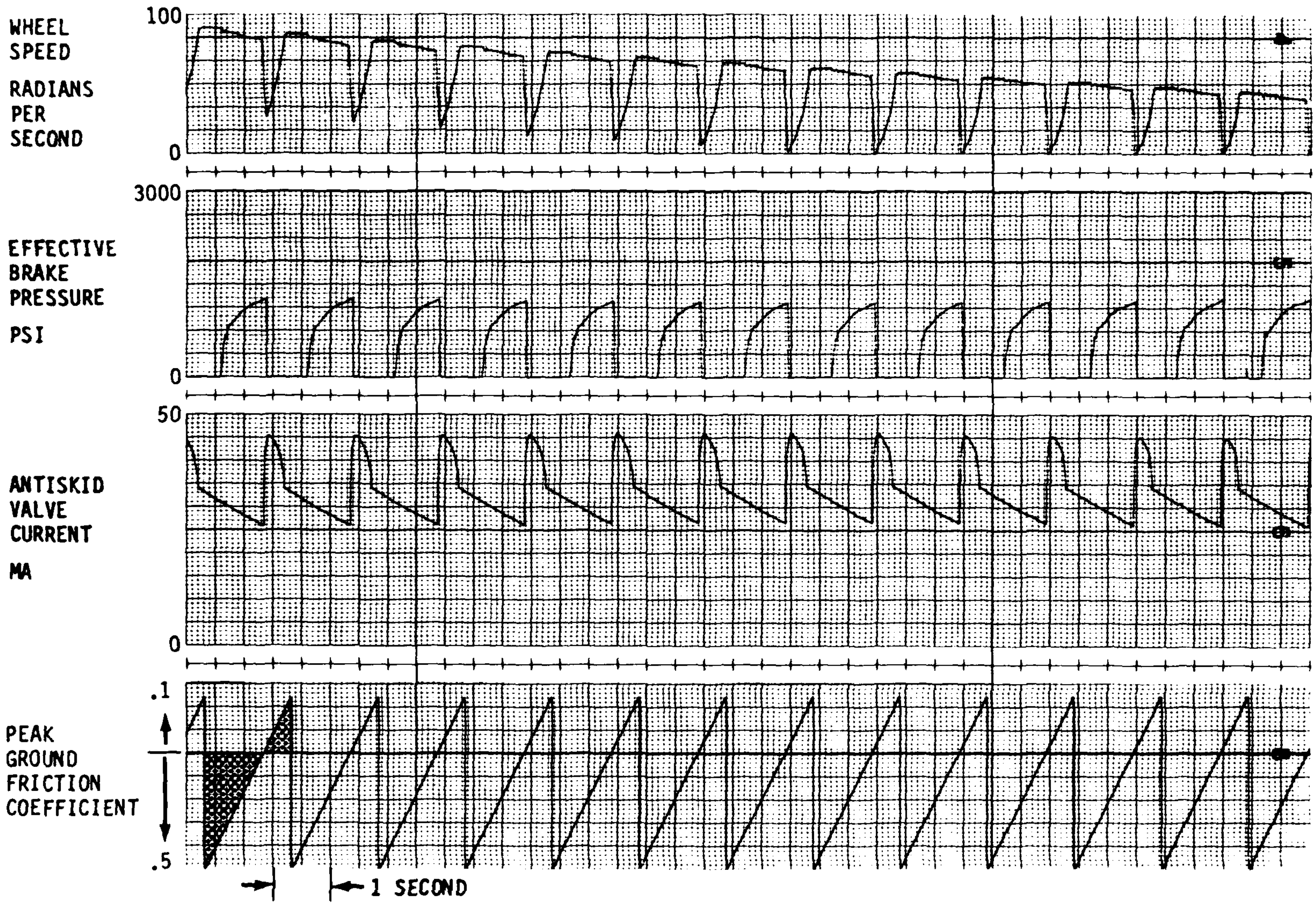


Figure E-40 Step Friction Performance, Standard System, +160°F

TABLE E-6 LANDING GEAR SYSTEM STABILITY, STANDARD SYSTEM

TEST CONDITION		COMMENTS
TEMPERATURE	FORE-AFT DOF STRUT DAMPING RATIO	
AMBIENT	.1 (NORMAL)	Strut oscillations are damped
	0	Strut oscillations are damped, strut oscillation amplitude is increased, strut oscillation superimposes ripple on wheel speed, small spikes in antiskid valve current indicates antiskid control box is interpreting strut oscillation as skids
-40	.1 (NORMAL)	Strut oscillations are damped
	0	Strut oscillations are undamped at low speed, strut oscillation superimposes ripple on wheel speed, strut oscillations are not interpreted as skids
160	.1 (NORMAL)	Strut oscillations are damped
	0	Strut oscillations are undamped at low speed, strut oscillation superimposes ripple ono wheel speed, strut oscillations are interpreted as skids

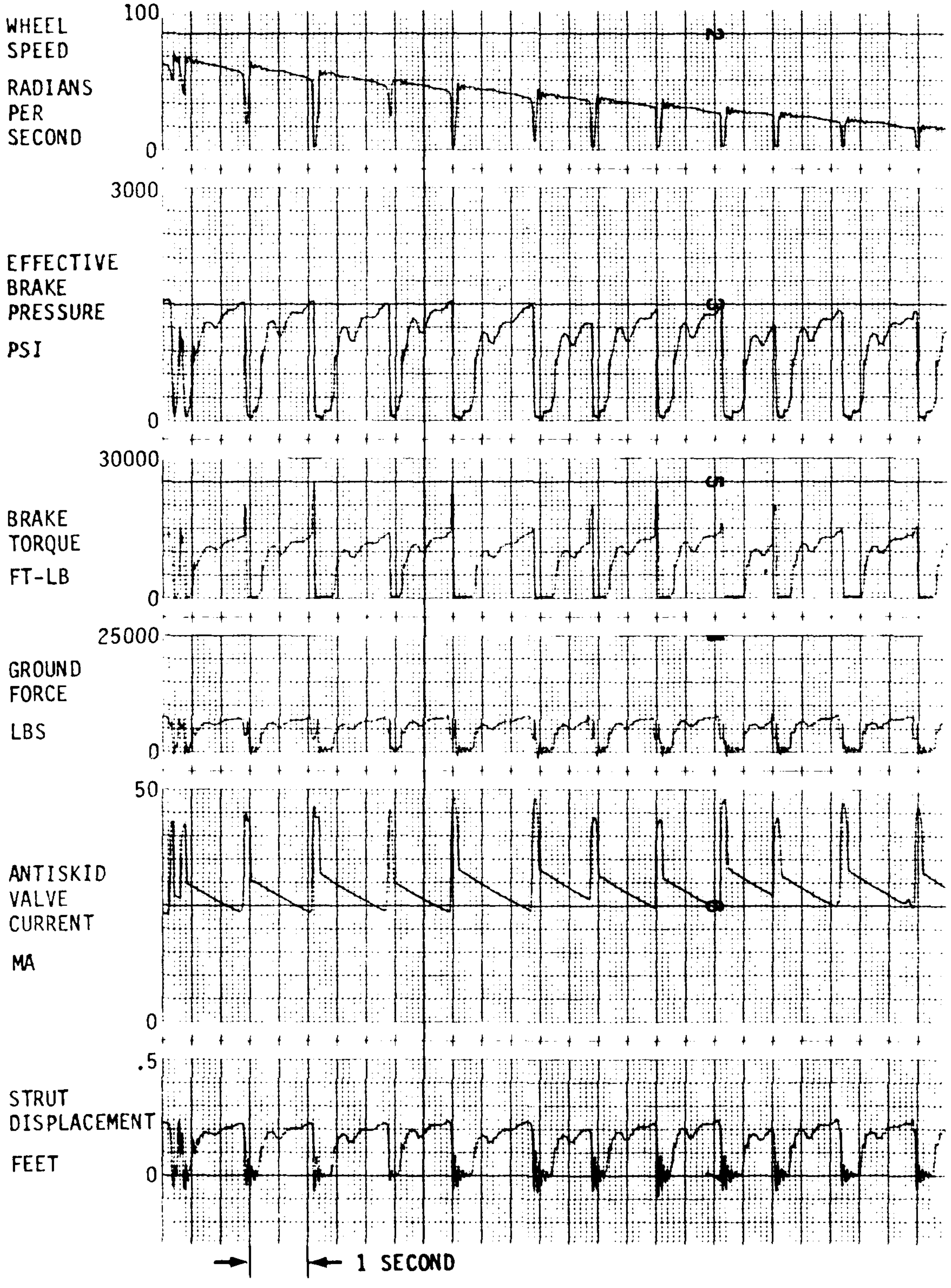


Figure E-41 Brake System Stability, Standard System, Normal Damping, Ambient

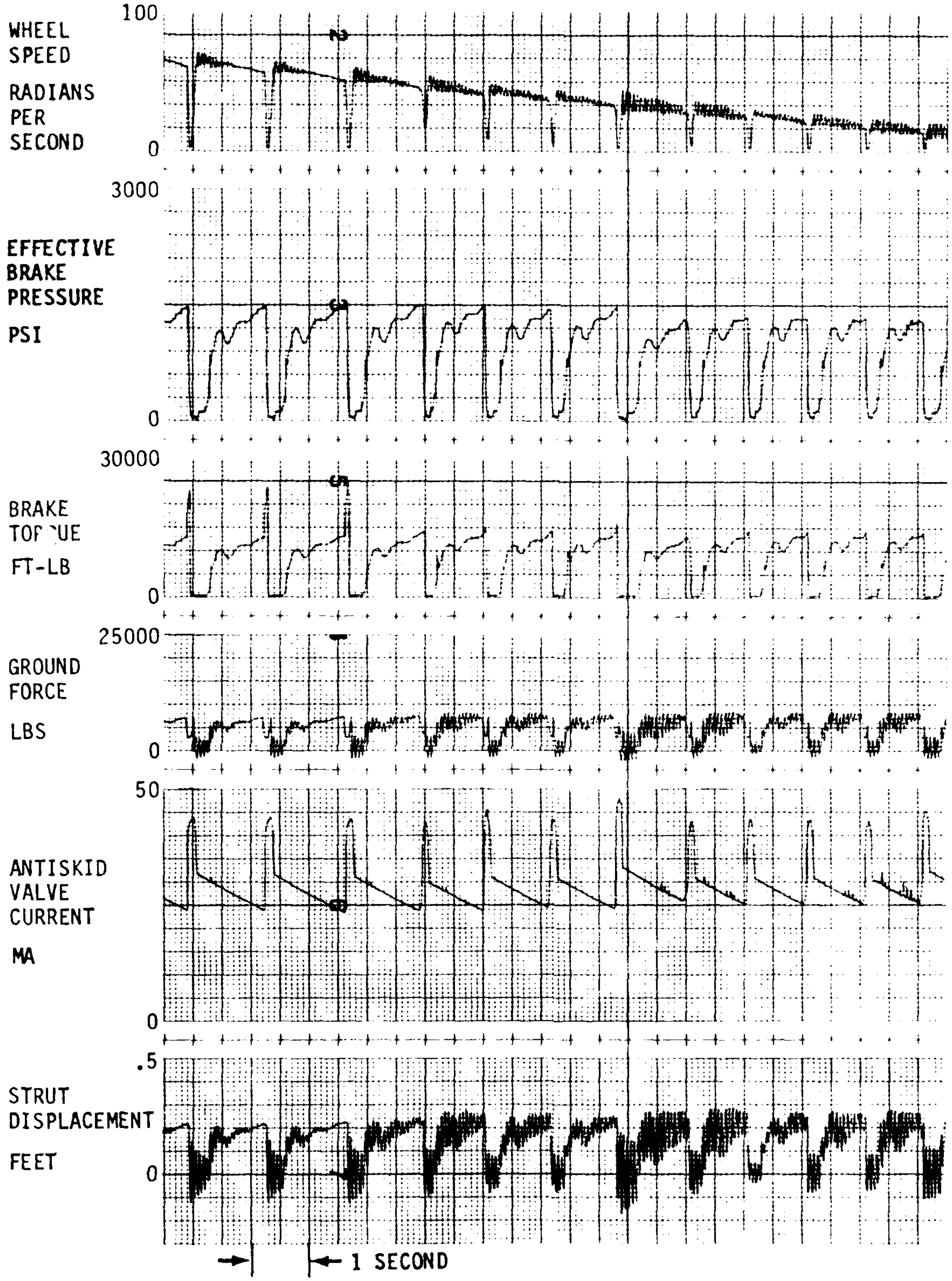


Figure E-42 Brake System Stability, Standard System, Zero Damping, Ambient

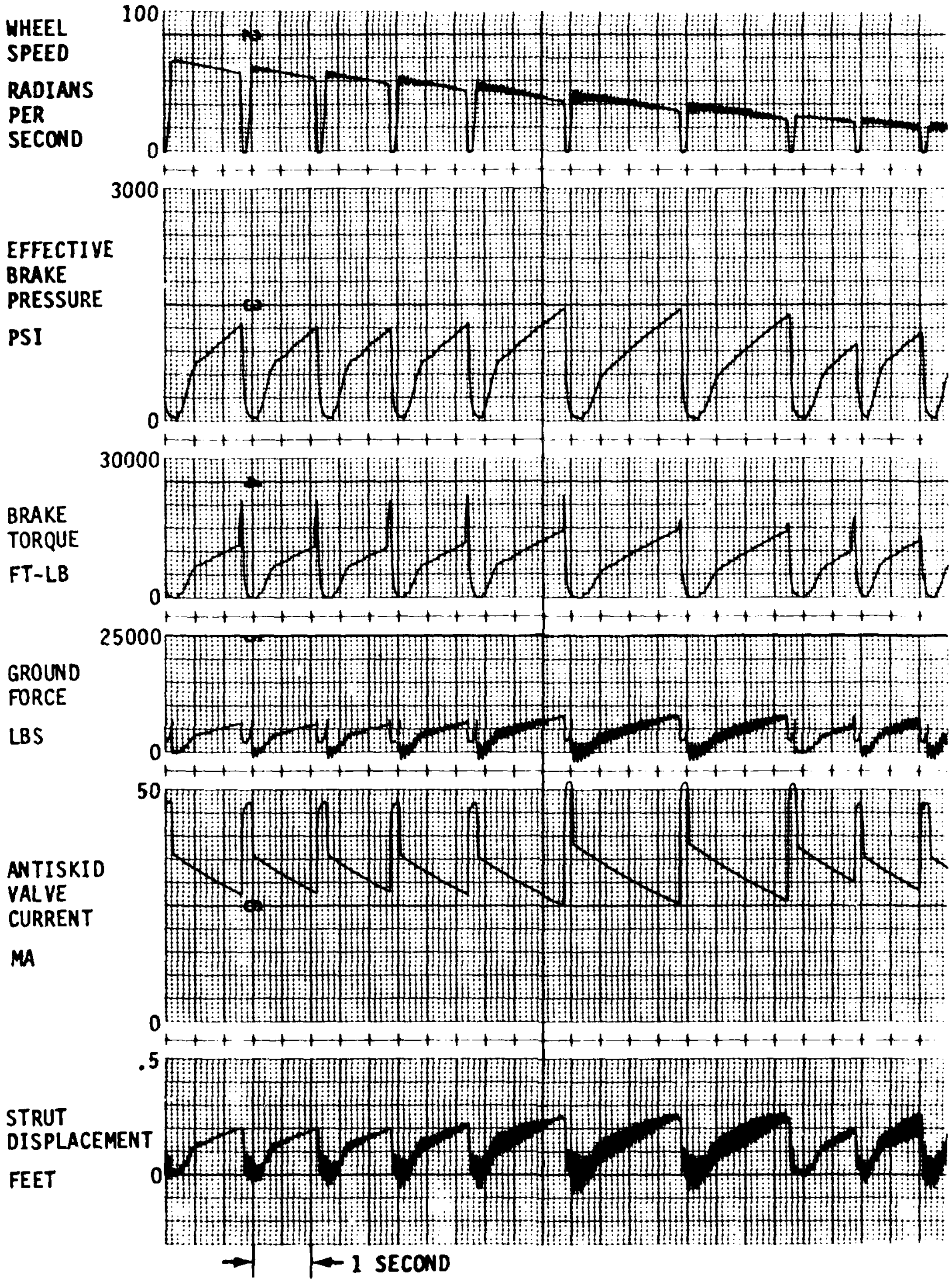


Figure E-43 **Brake System Stability, Standard System, Zero Damping, -40°F**

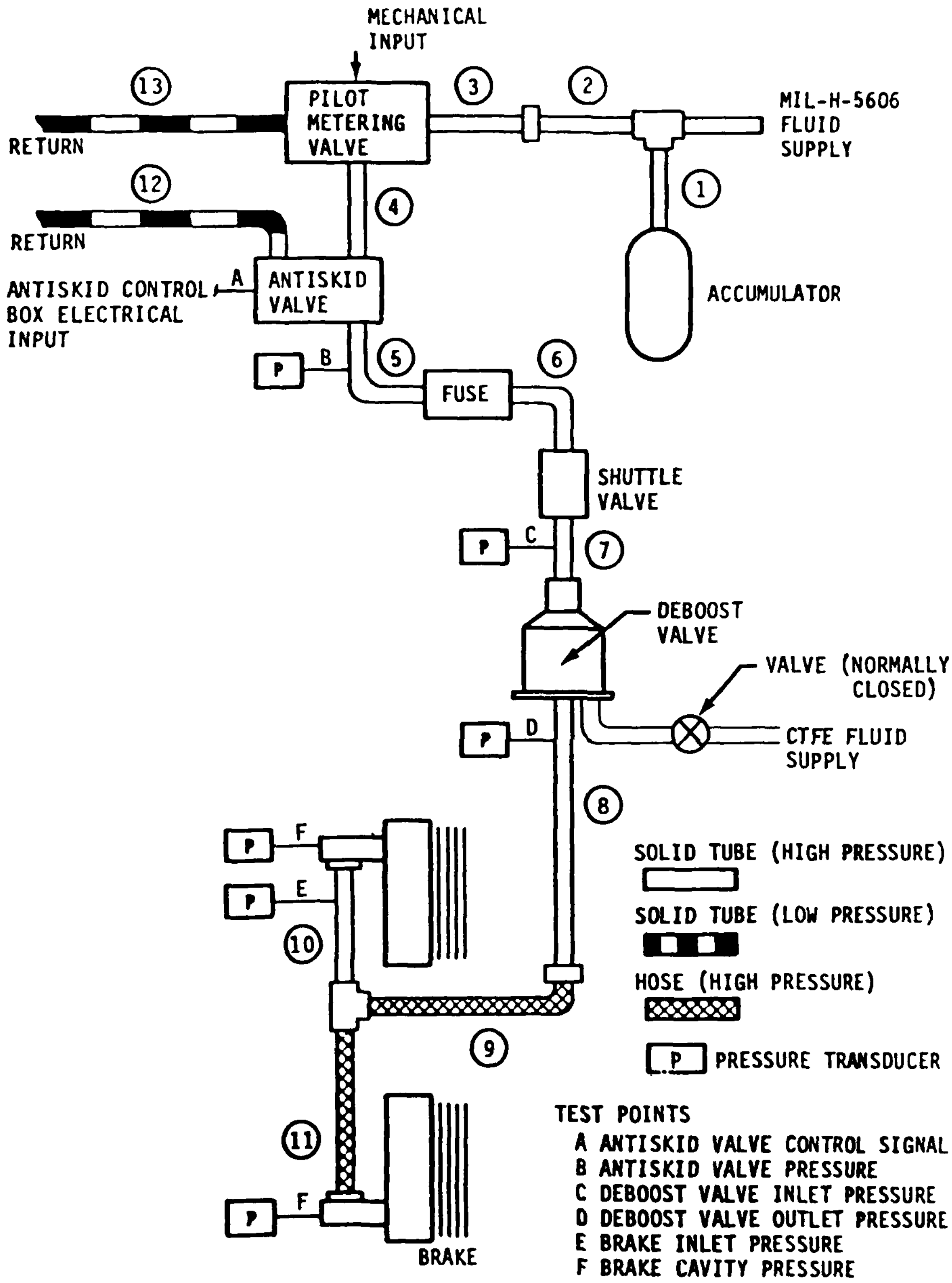


Figure E-44 Two-Fluid Brake Hydraulic System Schematic

pump containing CTFE fluid were included in the mockup for filling and servicing the CTFE portion of the hydraulic system. The mockup was instrumented as shown in Figure E-44. The location of the instrumentation points duplicated the location of the instrumentation points on the standard KC-135 brake hydraulic system mockup.

The mockup was serviced with MIL-H-5606 and CTFE per the procedure described in Section 4.1.2 of the Interim Technical Report (Appendix A). The mockup was integrated with the KC-135 Mark II antiskid control unit and the hybrid computer to form the two-fluid brake system airplane simulation.

The two-fluid brake system performance tests were performed to determine the dynamic response and braking performance of the two-fluid brake system. The test results were then compared to the baseline results to determine the effect which the two-fluid hydraulic brake system has upon airplane braking performance.

The system tests were performed at ambient, -40 degrees Fahrenheit and +160 degrees Fahrenheit. Low temperature testing at -65 degrees Fahrenheit was performed; however, continuous leakage from the antiskid valve occurred throughout the tests. See Section 3.4.4 for additional discussion of the leakage problem.

The ambient temperature tests were performed on April 15, 1981. The temperature in the test area was 70 degrees Fahrenheit.

The high temperature tests were performed on April 16, 1981. The brake hydraulic system was soaked for 7 hours and 5 minutes at 160 degrees Fahrenheit prior to the start of testing.

The low temperature tests at -40 degrees Fahrenheit were performed on April 17, 1981. The brake hydraulic system was soaked approximately 6 hours and 15 minutes at -40 degrees Fahrenheit prior to the start of testing.

The low temperature tests at -65 degrees Fahrenheit were performed on May 4, 1981. The brake hydraulic system was soaked for 6 hours and 45 minutes at -65 degrees Fahrenheit prior to the start of testing.

The results of each two-fluid brake system performance test are described below. The brief descriptions of the test objective and test procedure which were included in the baseline test results write up have been deleted. The reader may refer to the baseline results for these brief descriptions or to the test plan for a comprehensive description.

E.2.1 FREQUENCY RESPONSE, TEST 1

The frequency response (gain and phase angle) of the brake system, antiskid valve and deboost valve were determined at ambient, +160 degrees Fahrenheit, -40 degrees Fahrenheit and -65 degrees Fahrenheit. The frequency response test conditions are given in Table E-3. The test results are given in Figures E-45 thru E-56.

E.2.2 STEP RESPONSE, TEST 2

The dynamic response of the two-fluid brake hydraulic system to a step change in the commanded brake pressure level was measured a several locations in the system. The tests were performed at ambient, +160 degrees Fahrenheit, -40 degrees Fahrenheit and -65 degrees Fahrenheit. The step response conditions and test points are given in Table E-4. The pressure step results are given in Figures E-57 thru E-64.

E.2.3 STATIC ANTISKID VALVE CURRENT VERSUS BRAKE PRESSURE, TEST 3

The antiskid valve used in the two-fluid brake system mockup was the same unit as used in the standard KC-135 brake system mockup. The current pressure characteristic of the vavle was determined during the baseline system performance tests. The reader is referred to Section E.1.3 for the results of the test.

E.2.4 STATIC BRAKE PRESSURE VERSUS BRAKE VOLUME, TEST 4

The pressure volume characteristic of each brake used in the two-fluid brake hydraulic system mockup was measured. The test results are given in Figure E-65.

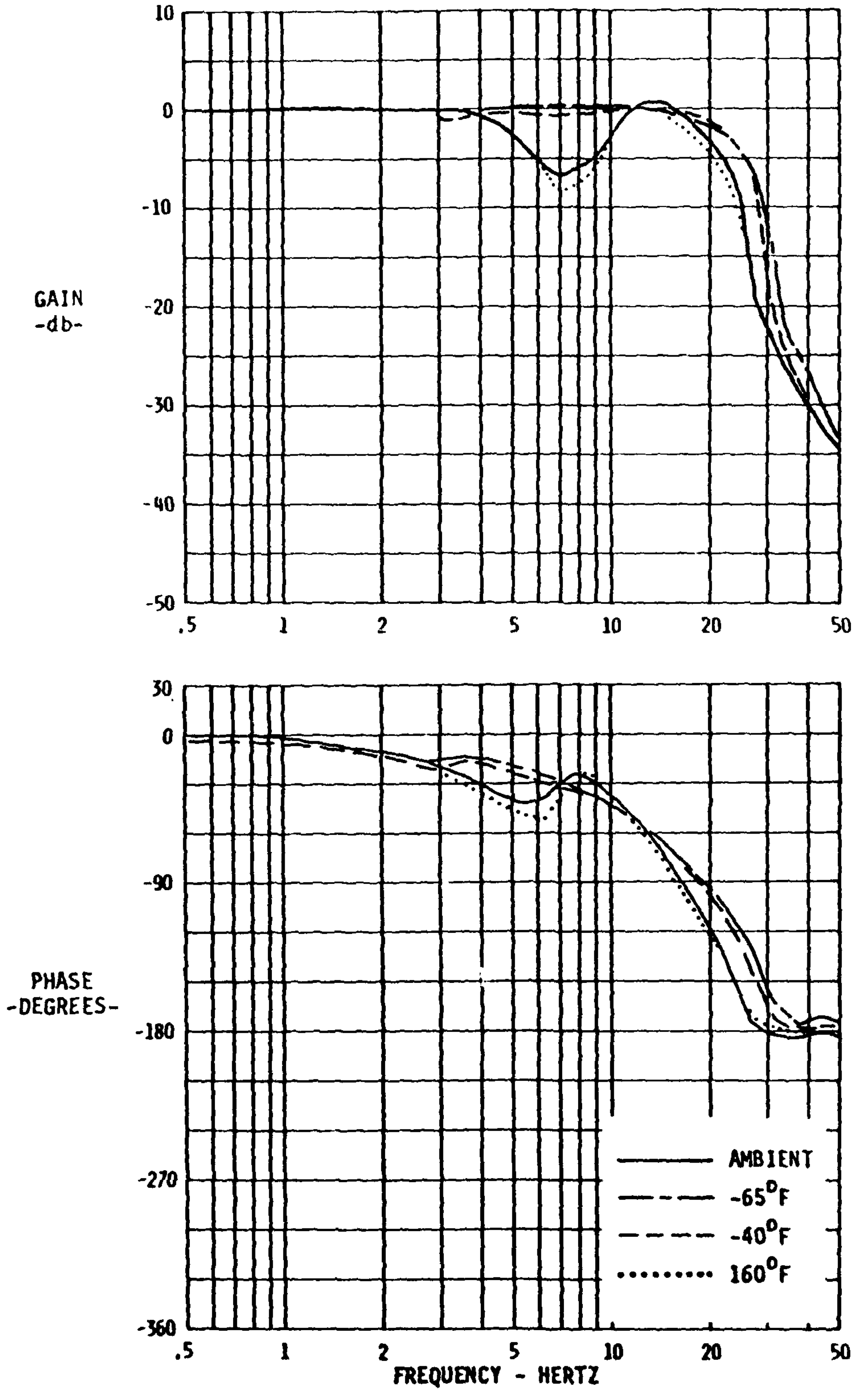


Figure E-45 Frequency Response, Two-Fluid System Antiskid Valve, 33% ± 100 psi

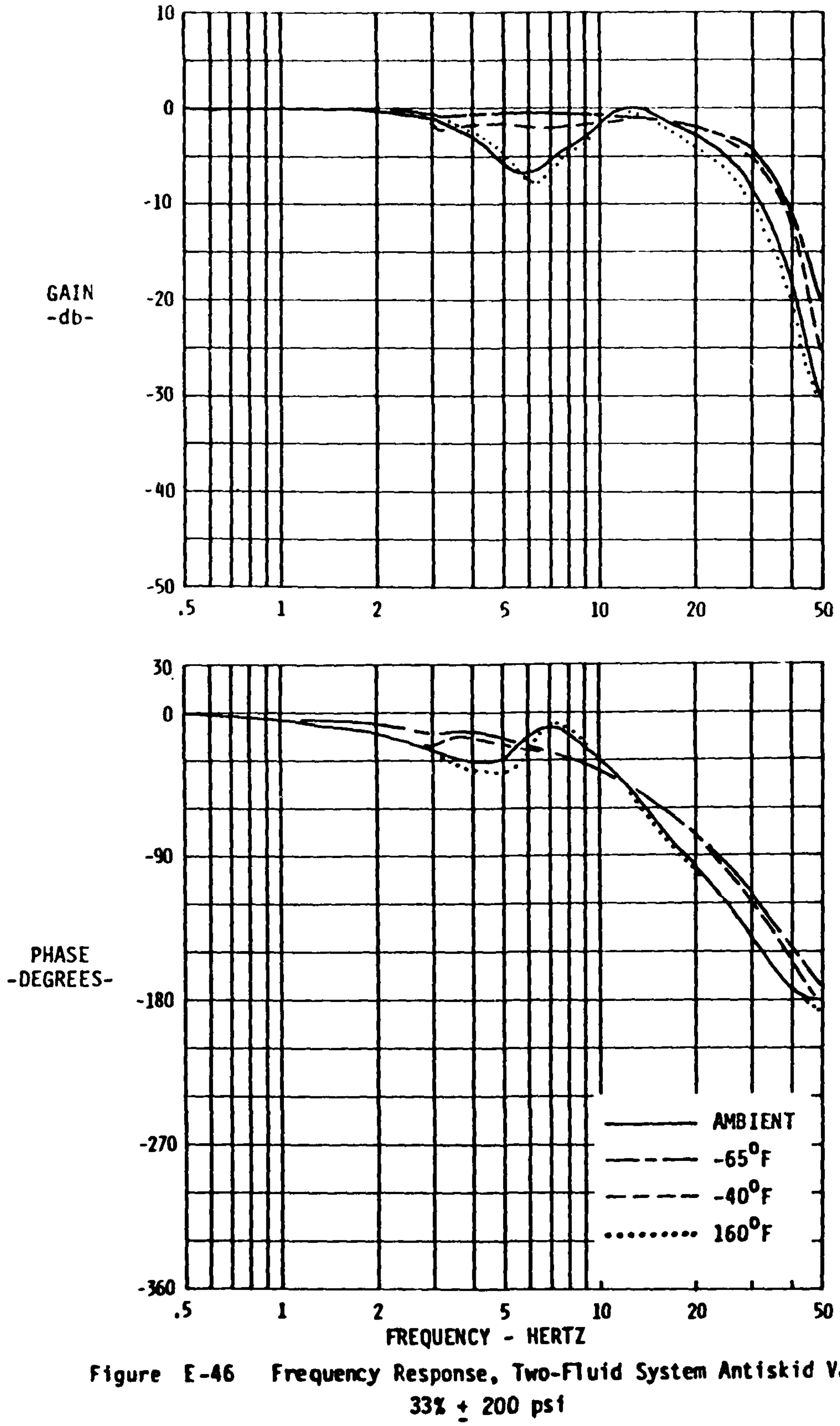


Figure E-46 Frequency Response, Two-Fluid System Antiskid Valve, 33% ± 200 psi

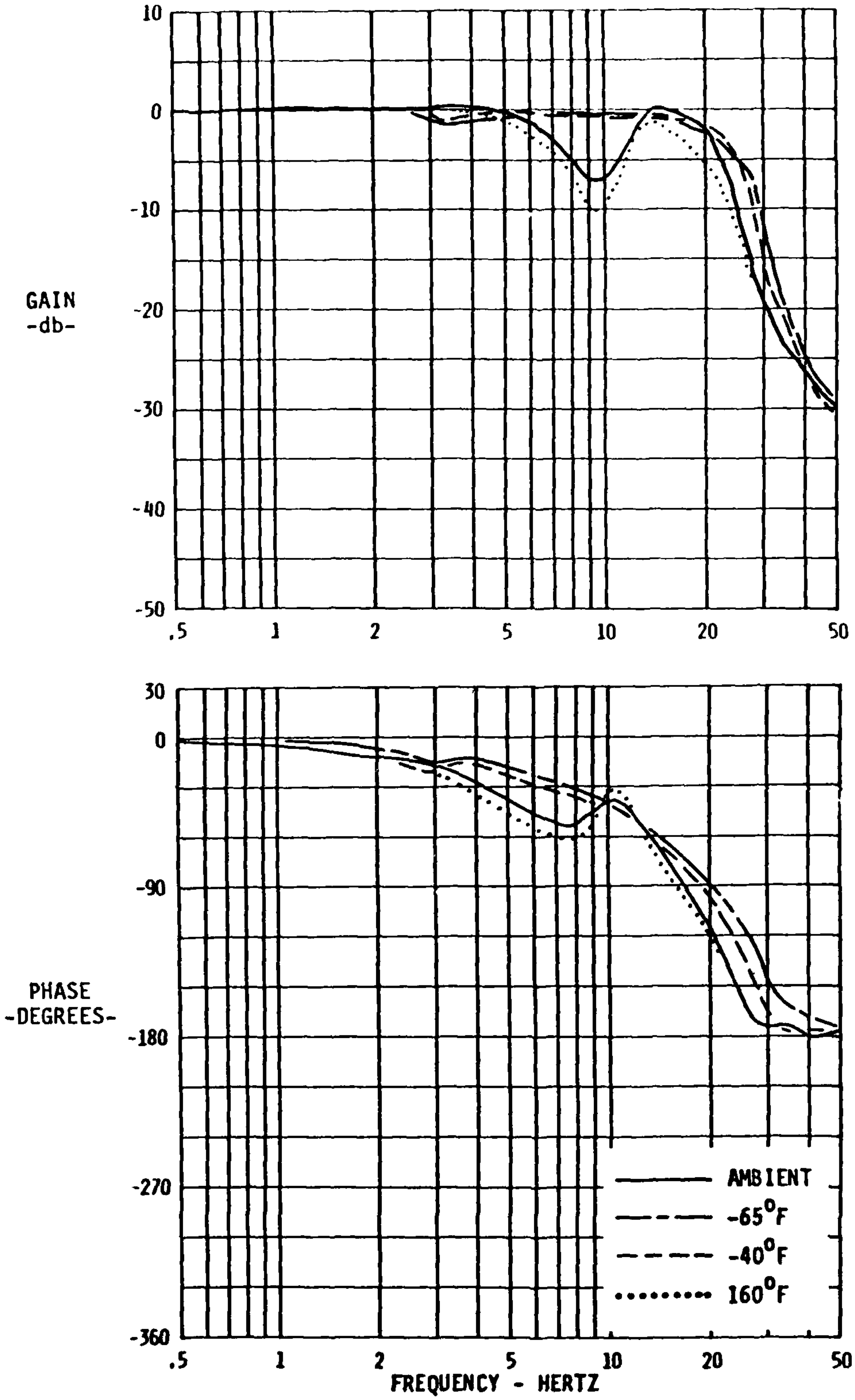


Figure E-47 Frequency Response, Two-Fluid System Antiskid Valve, 66% ± 100 psi

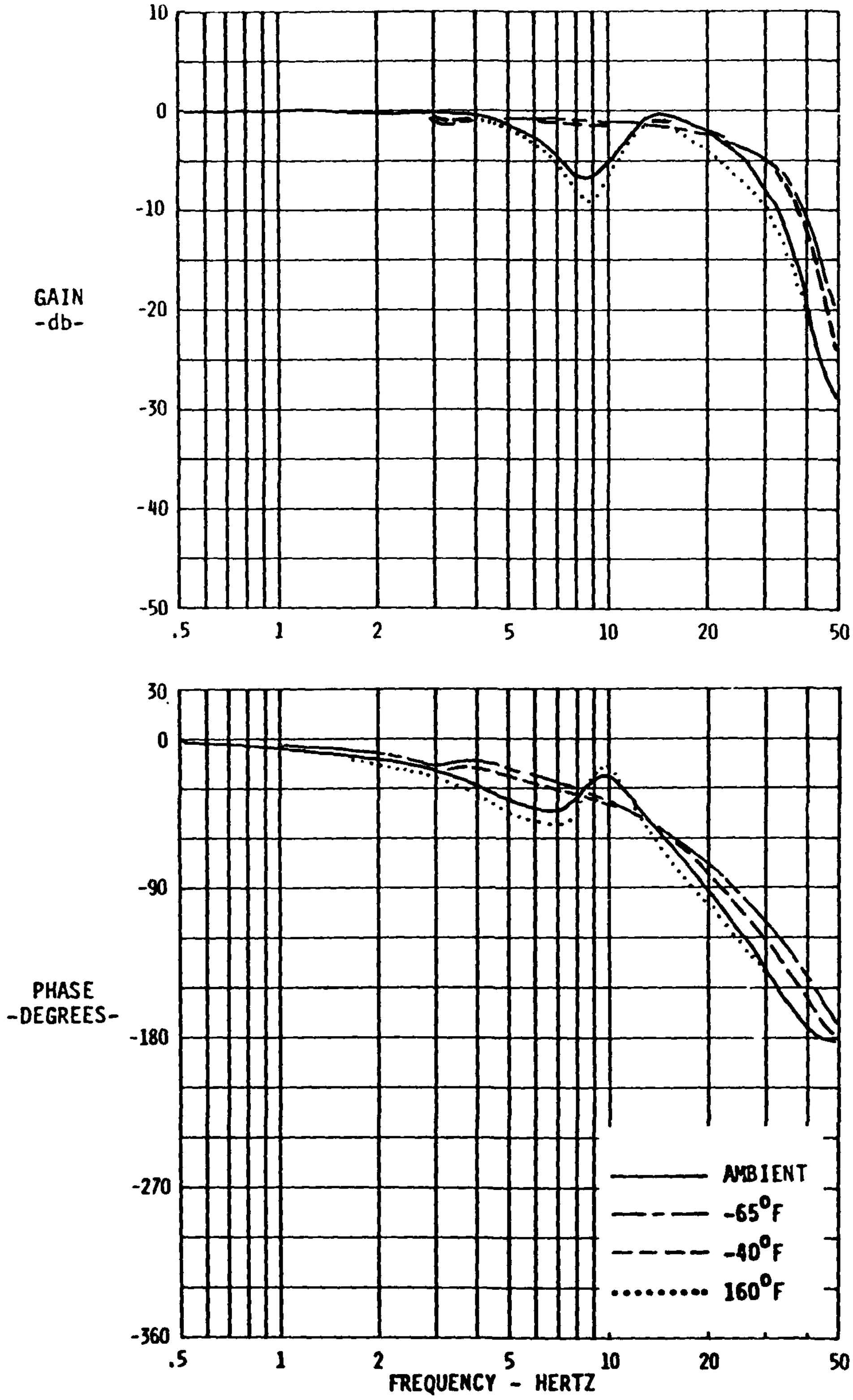


Figure E-48 Frequency Response, Two-Fluid System Antiskid Valve, 66% ± 200 psi

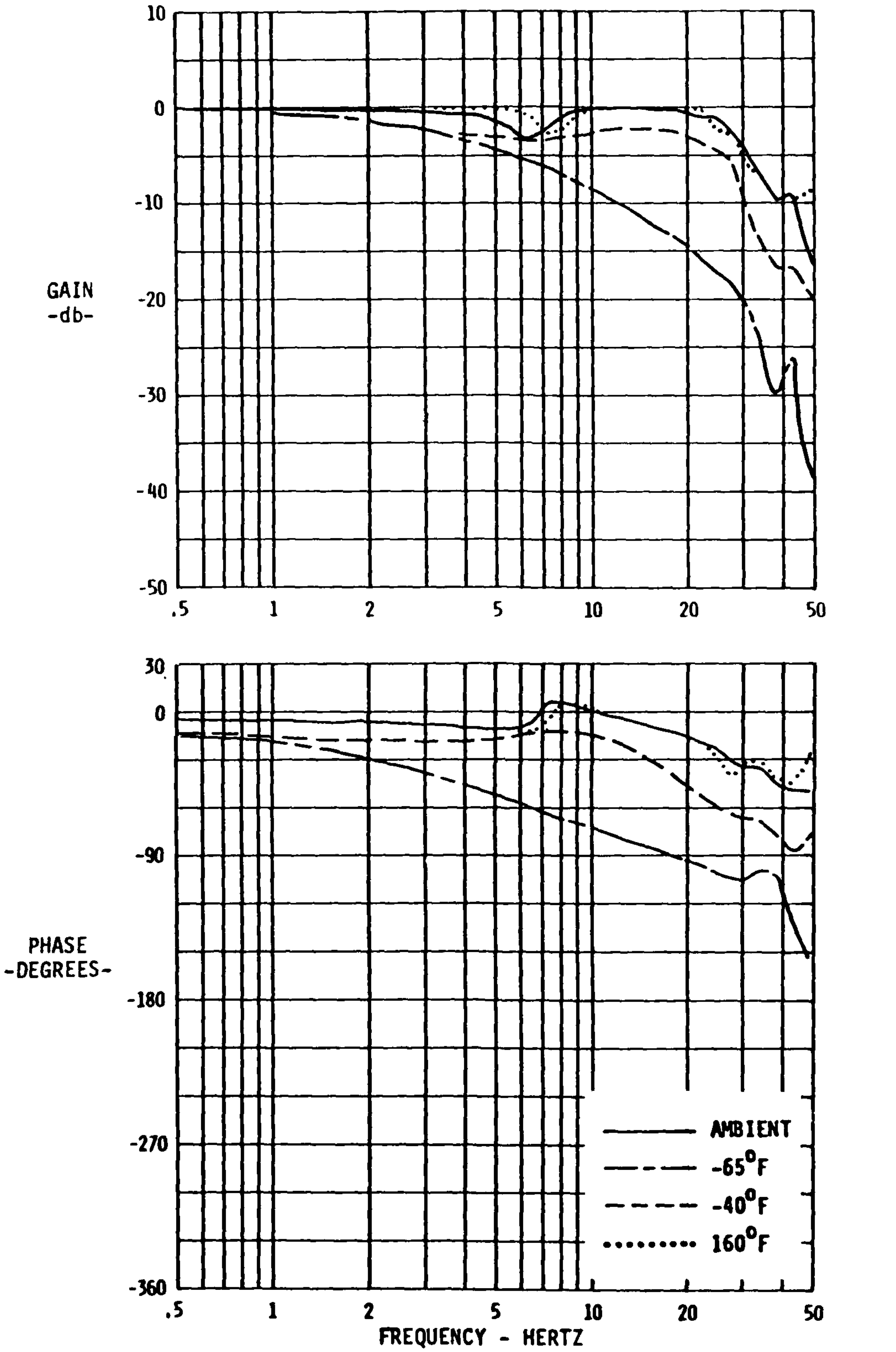


Figure E-49 Frequency Response, Two-Fluid Deboost Valve, 33% ± 100 psi

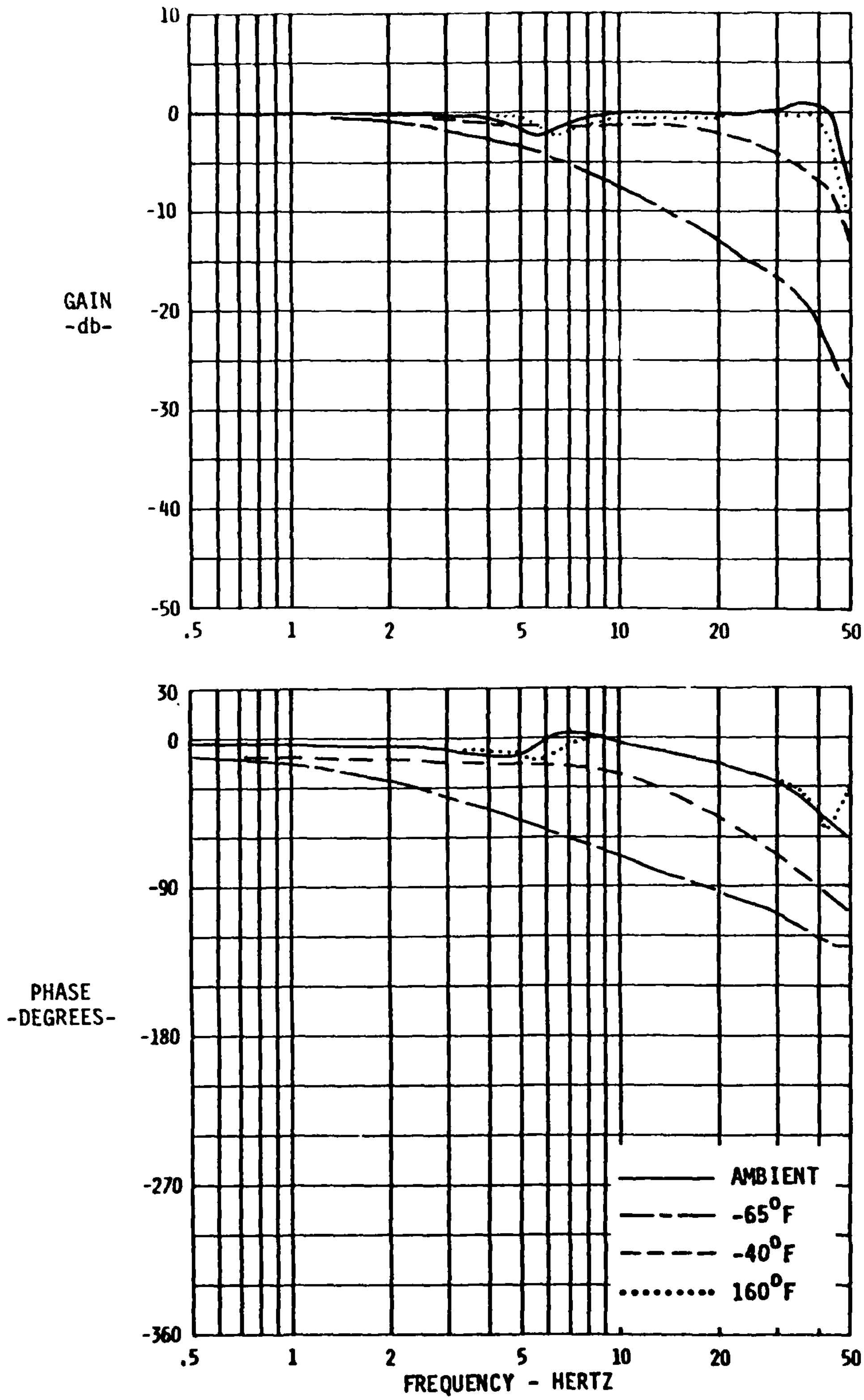


Figure E-50 Frequency Response, Two-Fluid Deboost Valve, 33% ± 200 psi

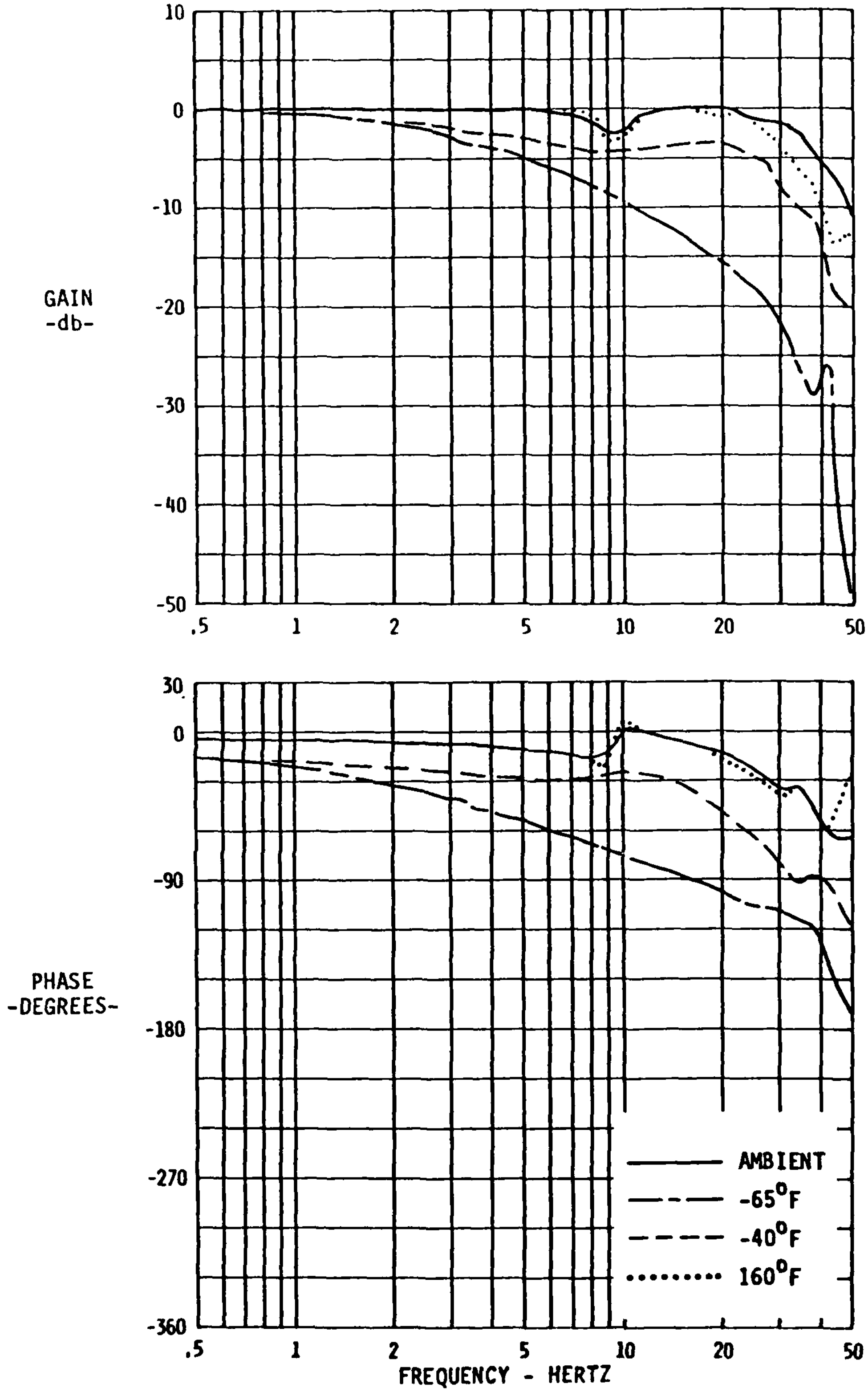


Figure E-51 Frequency Response, Two-Fluid Deboost Valve, 66% ± 100 psi

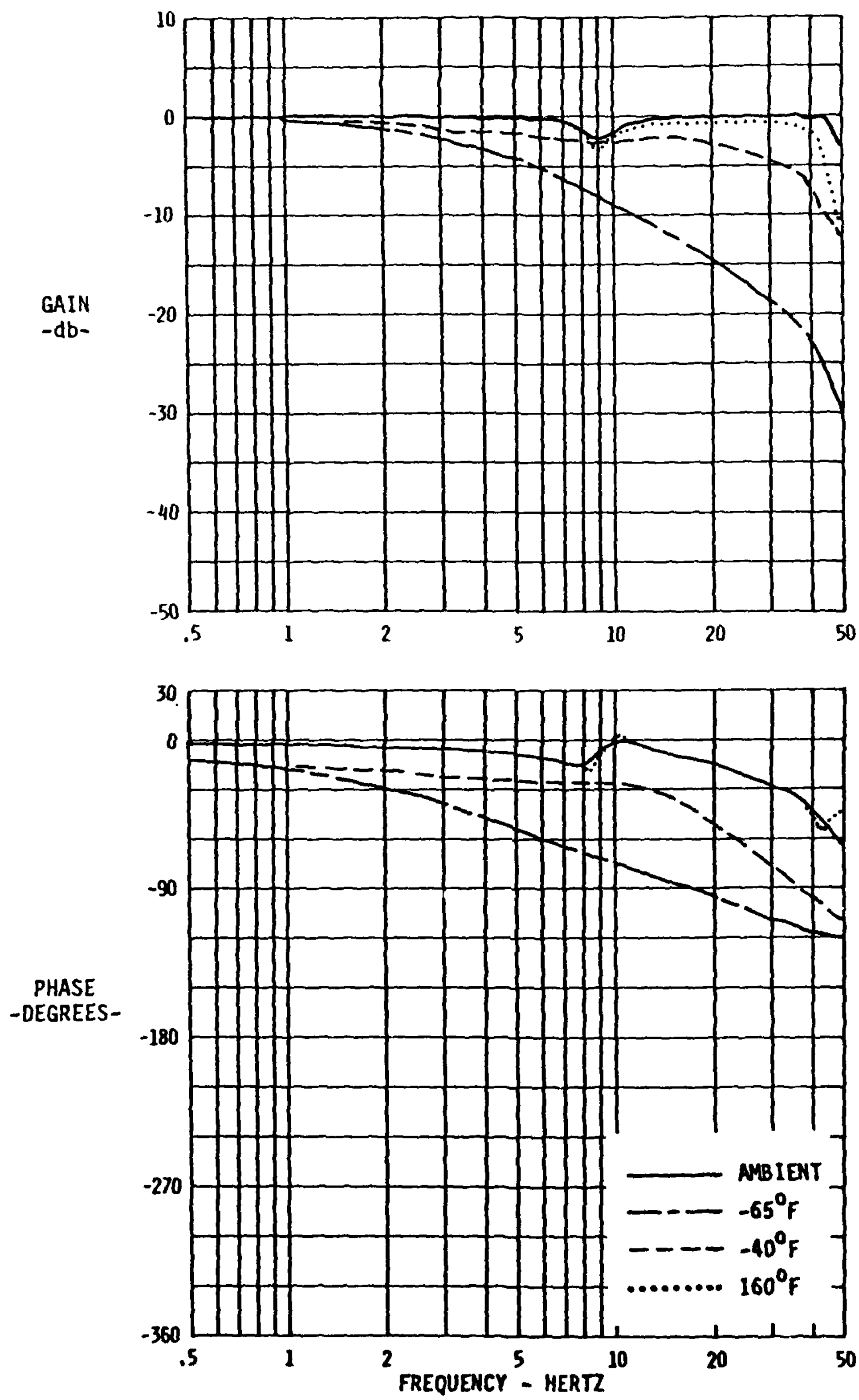


Figure E-52 Frequency Response, Two-Fluid Deboost Valve, 66% ± 200 psi

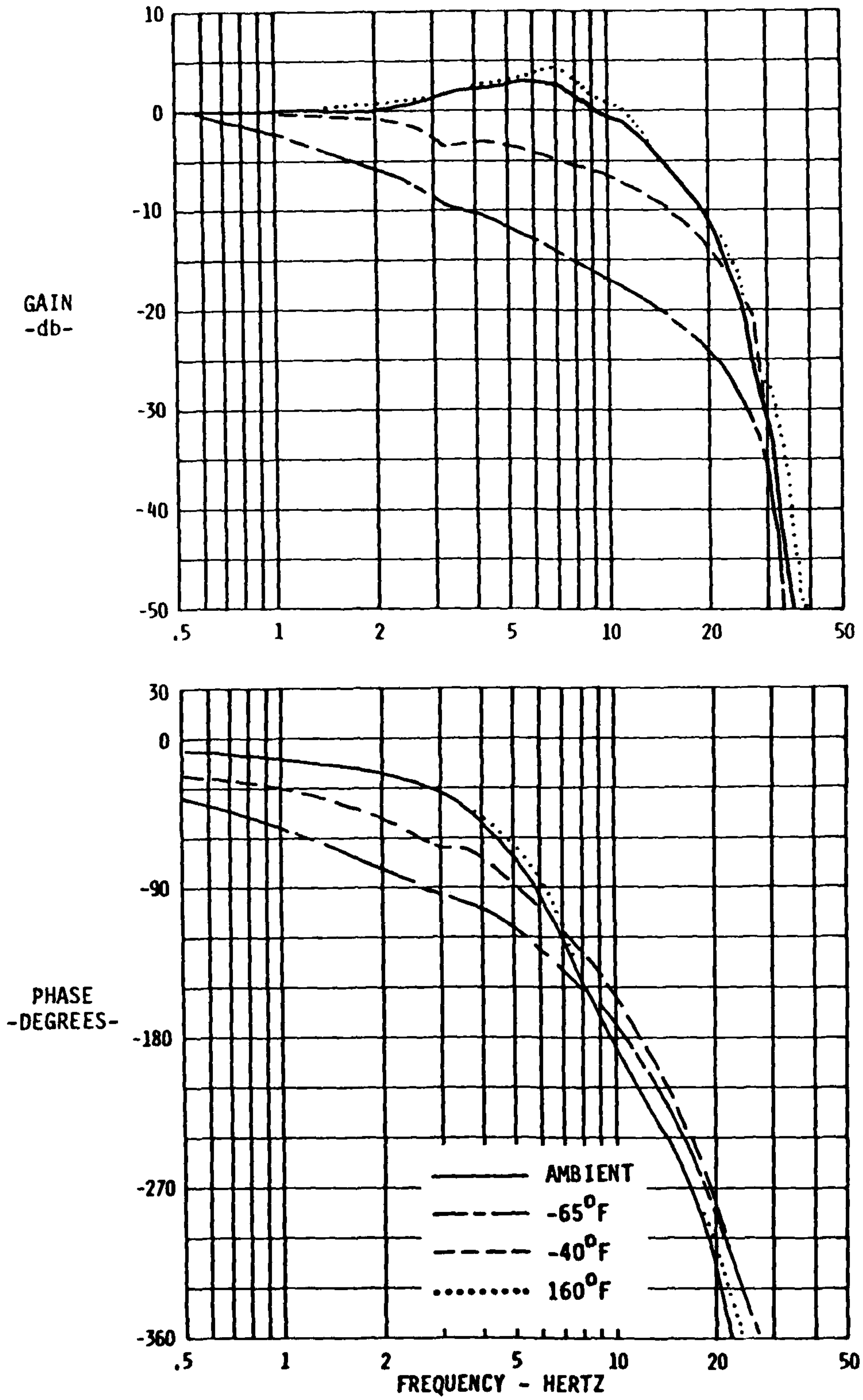


Figure E-53 Frequency Response, Two-Fluid Brake System, 33% ± 100 psi

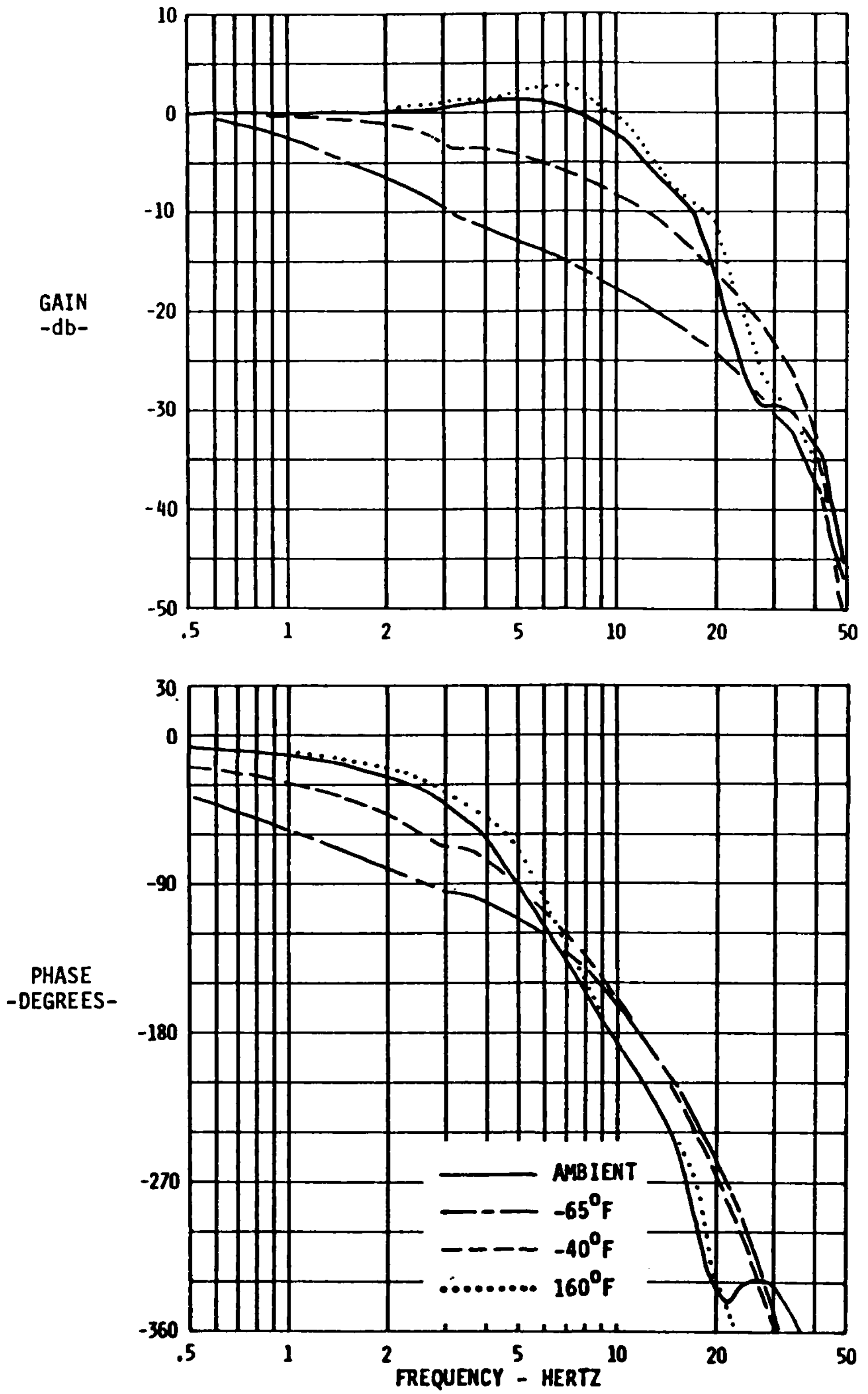


Figure E-54 Frequency Response, Two-Fluid Brake System, 33% ± 200 psi

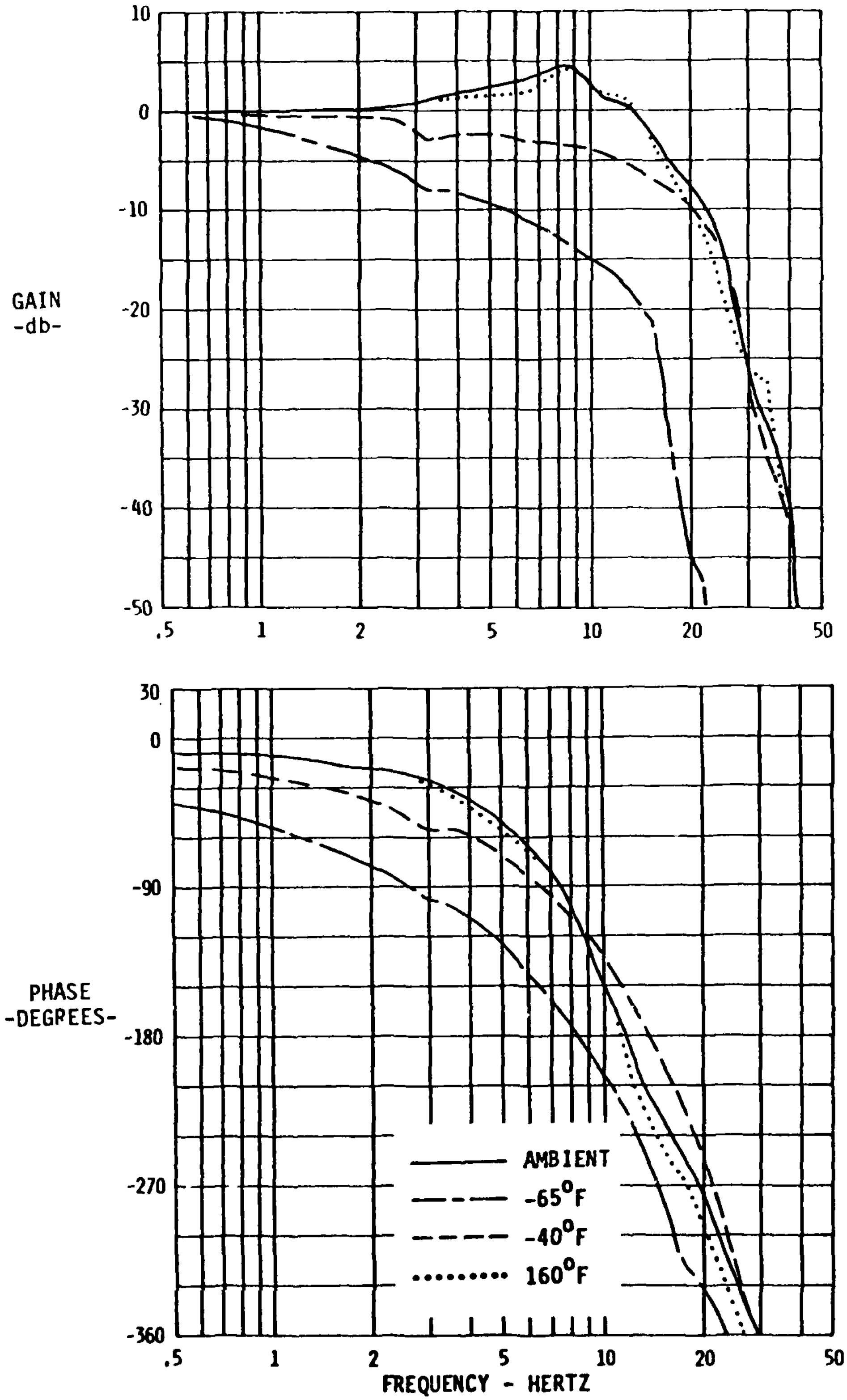


Figure E-55 Frequency Response, Two-Fluid Brake System, 66% ± 100 psi

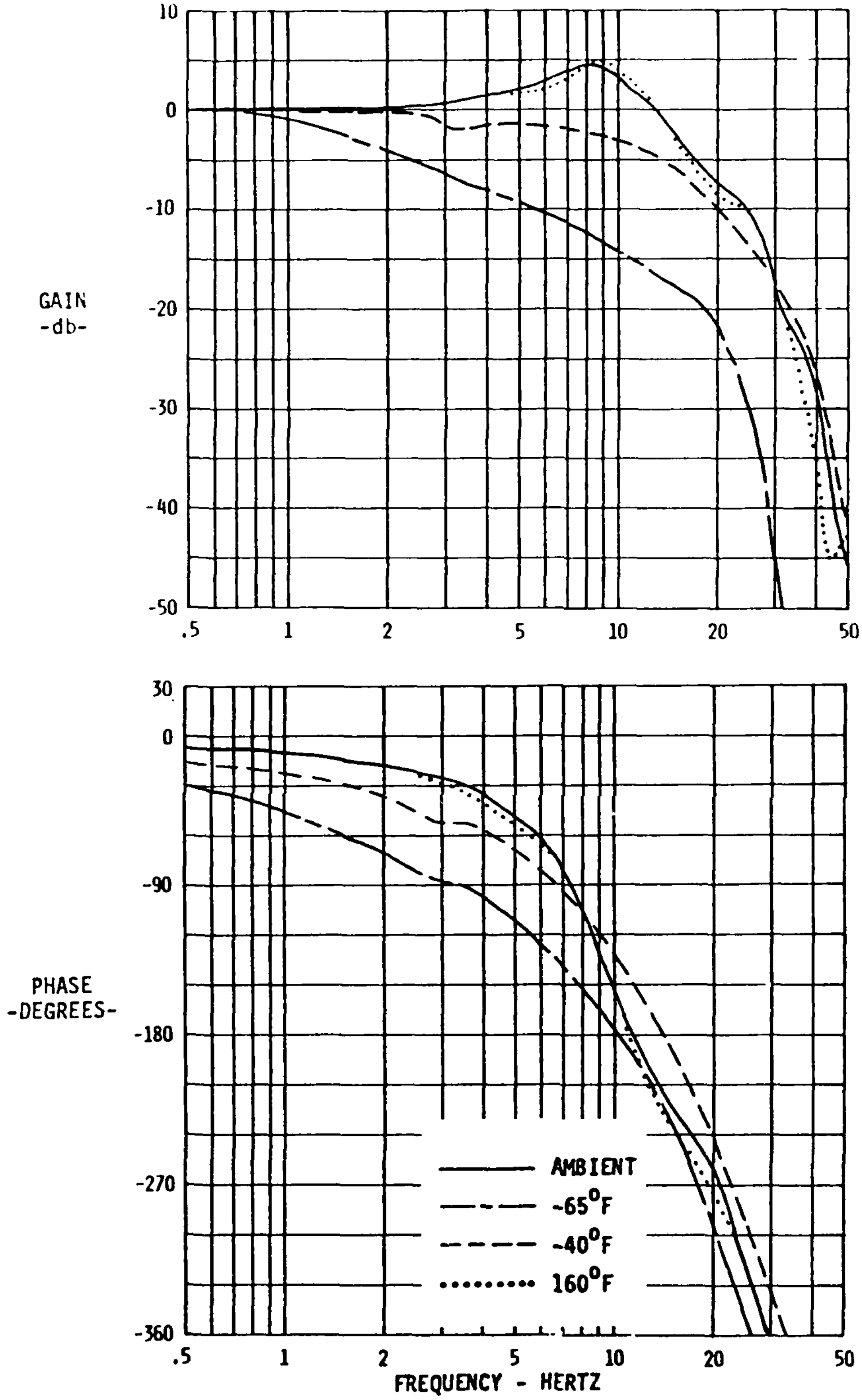


Figure E-56 Frequency Response, Two-Fluid Brake System, 66% ± 200 psi

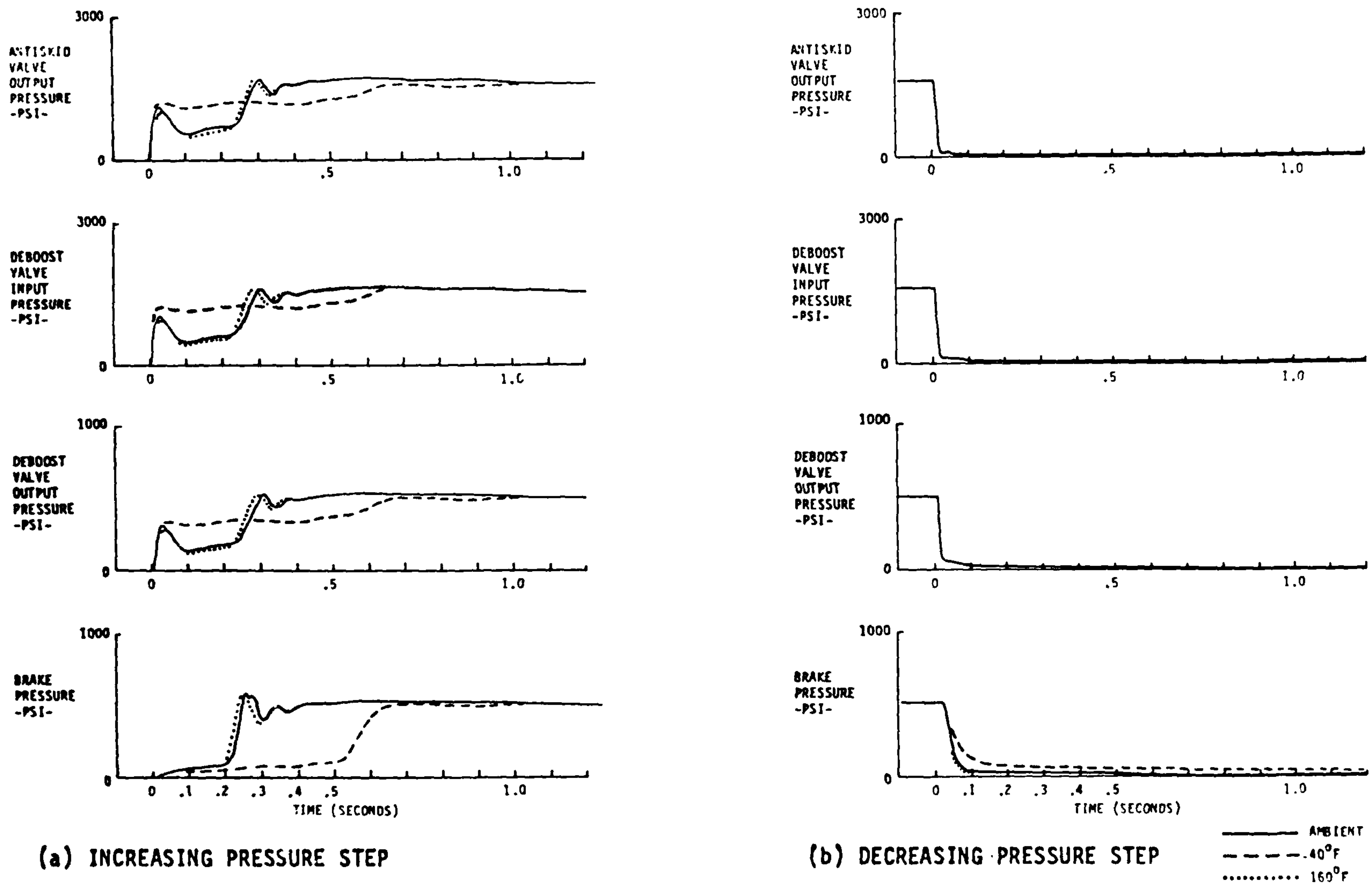


Figure E-57 Step Response, Two-Fluid System, 0-50%

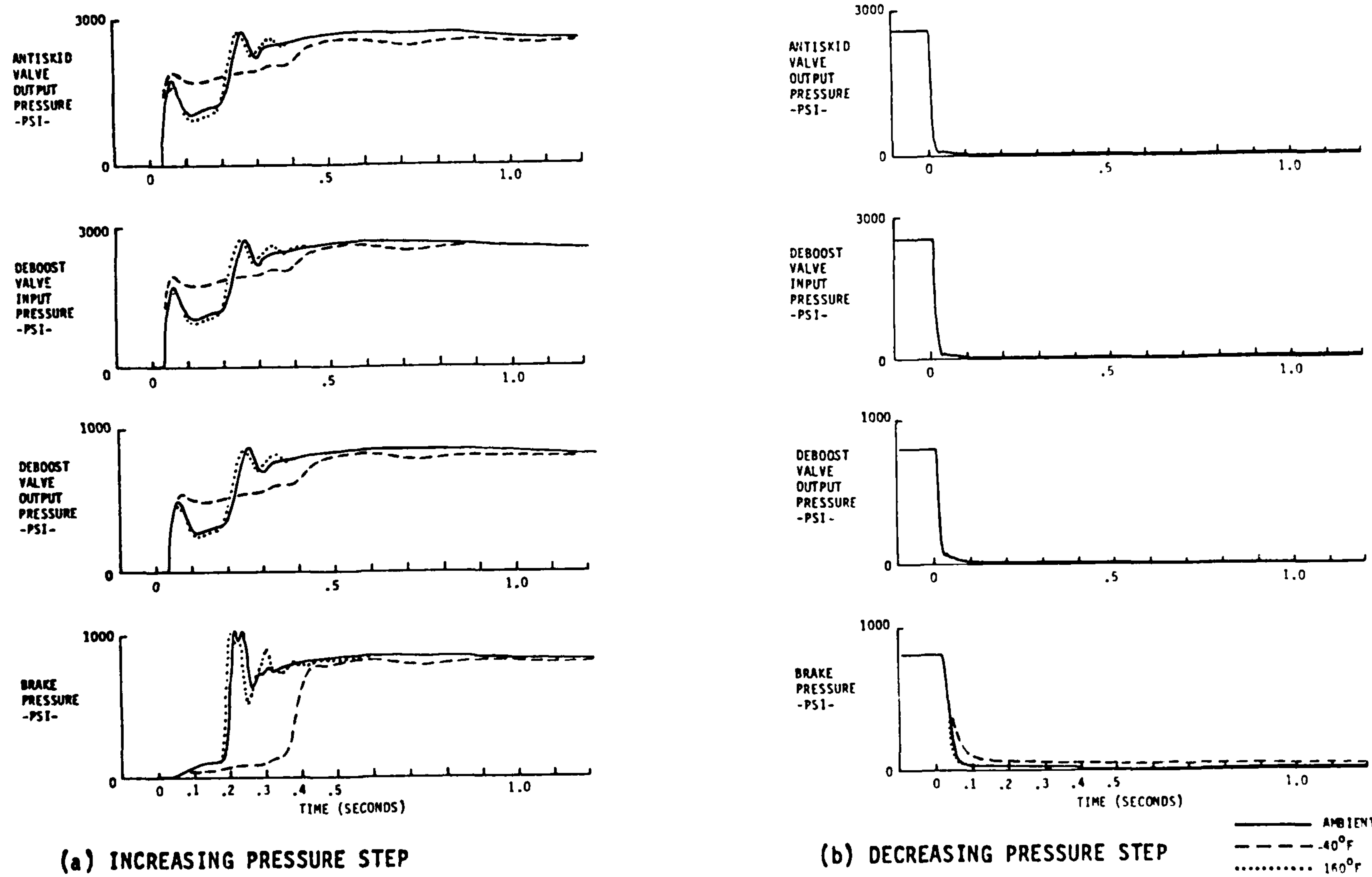
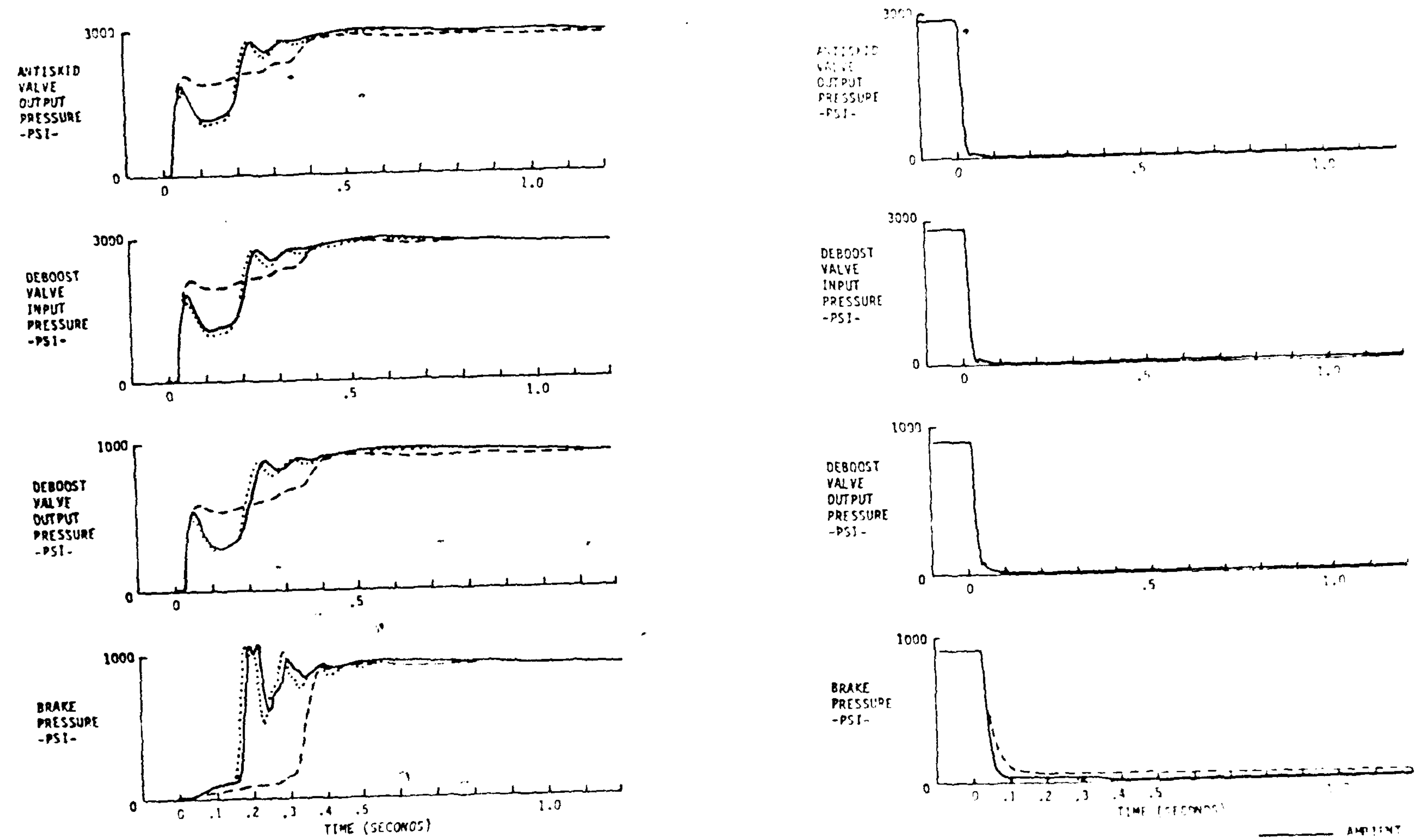


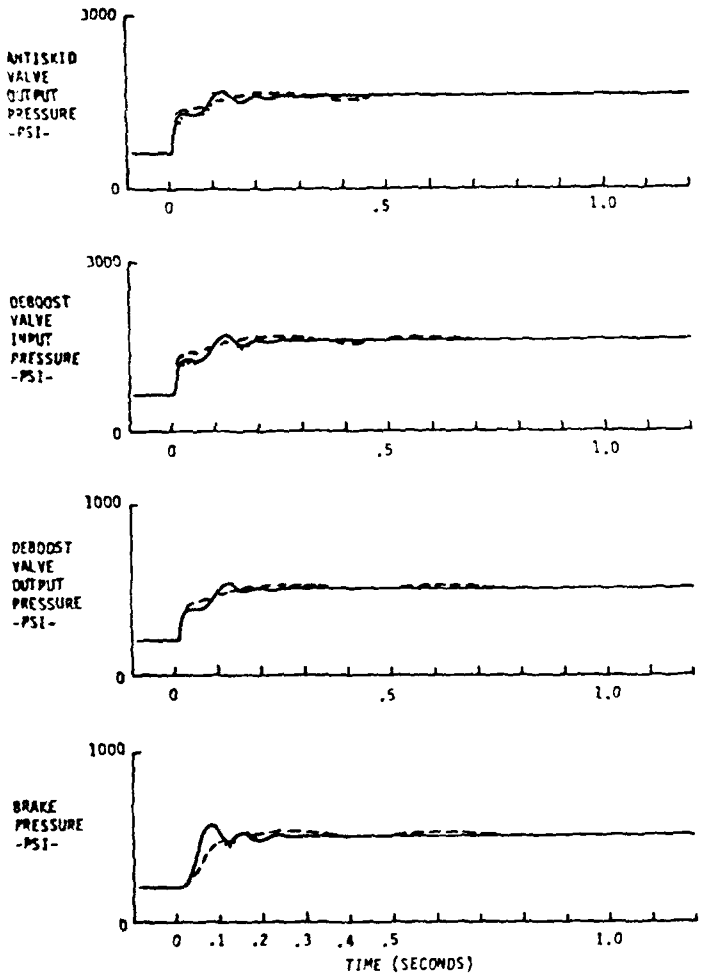
Figure E-58 Step Response, Two-Fluid System, 0-80%



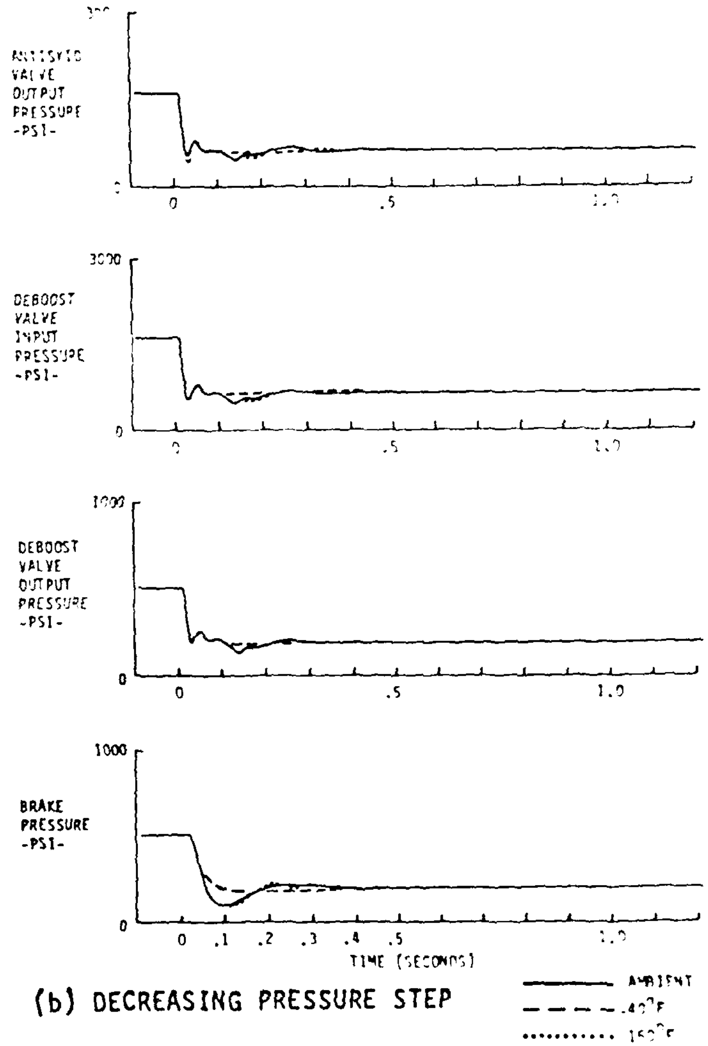
(a) INCREASING PRESSURE STEP

(b) DECREASING PRESSURE STEP

Figure E-59 Step Response, Two-Fluid System, 0-100%



(a) INCREASING PRESSURE STEP



(b) DECREASING PRESSURE STEP

Figure E-60 Step Response, Two-Fluid System, 20-50%

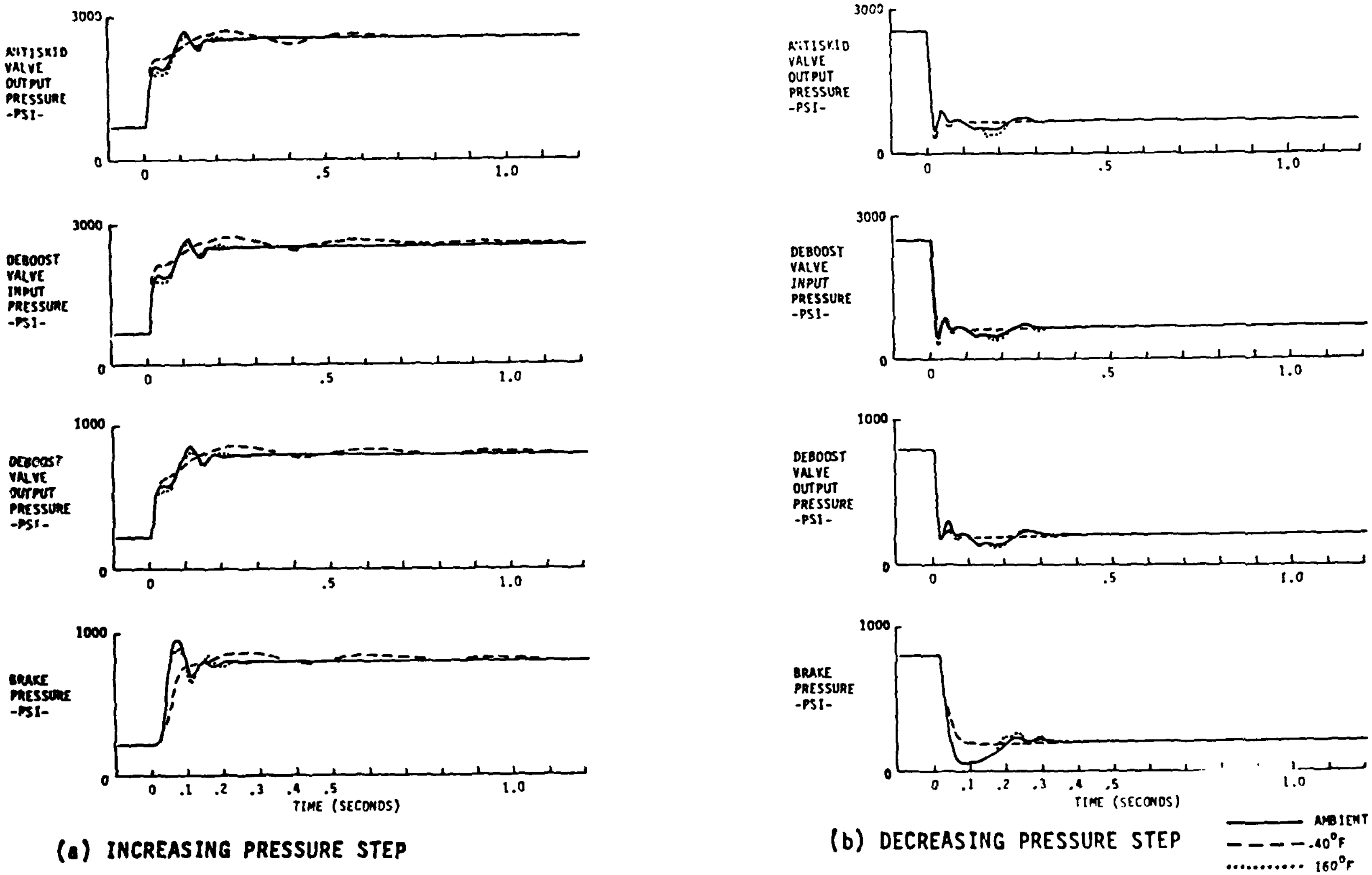


Figure E-61 Step Response, Two-Fluid System, 20-80%

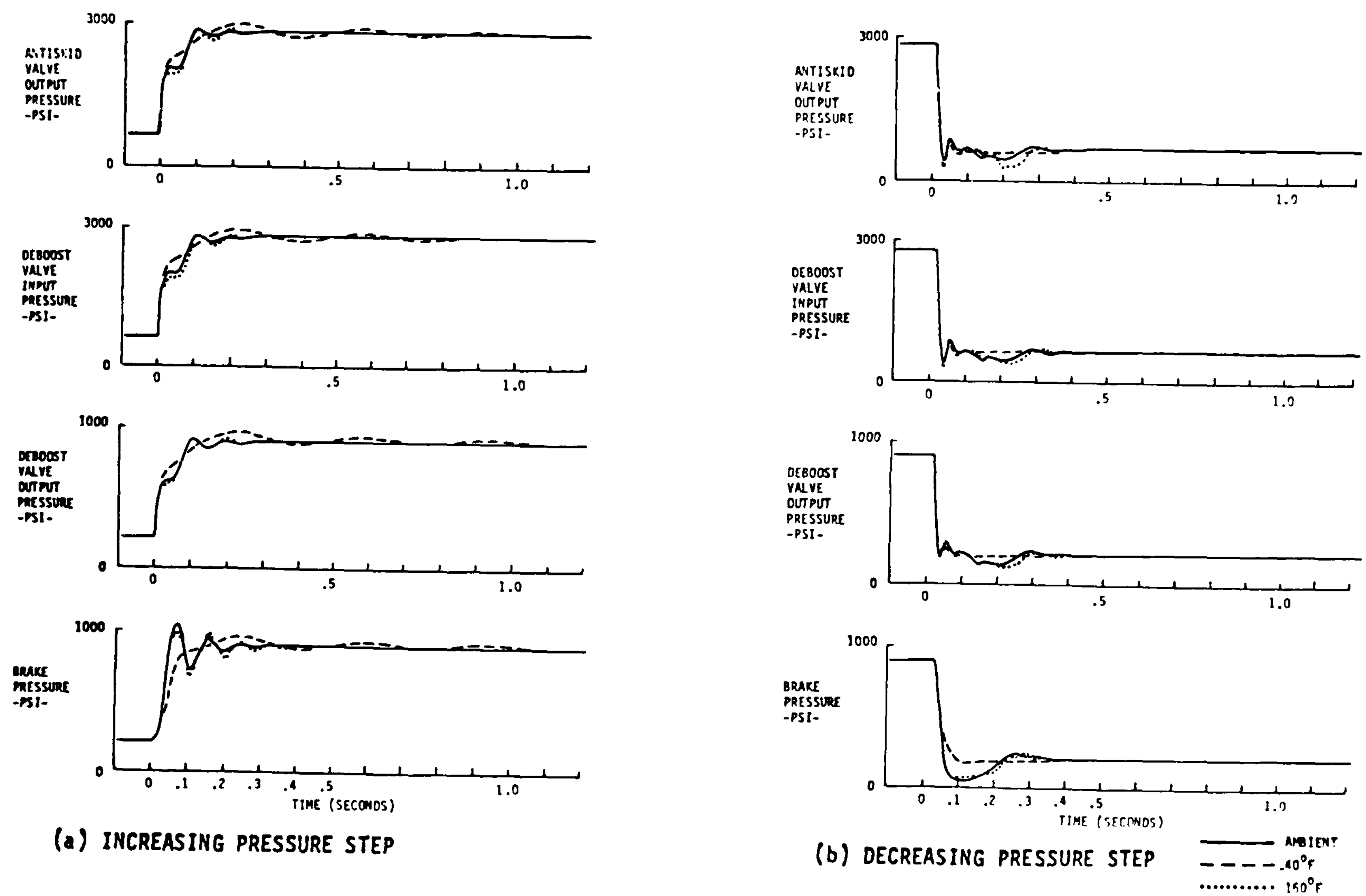
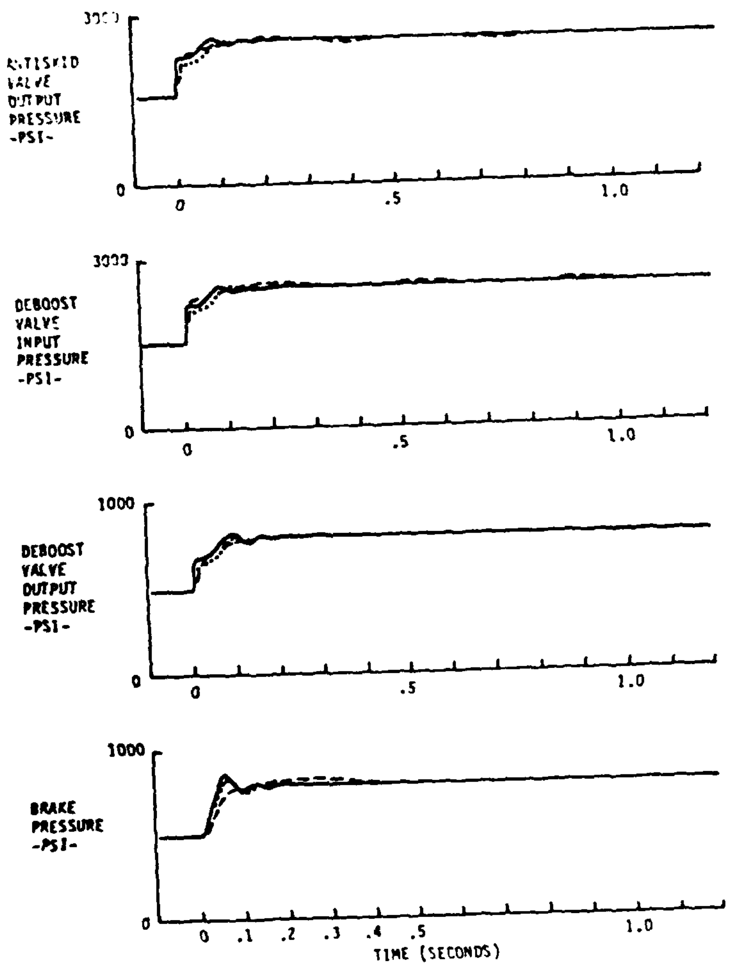
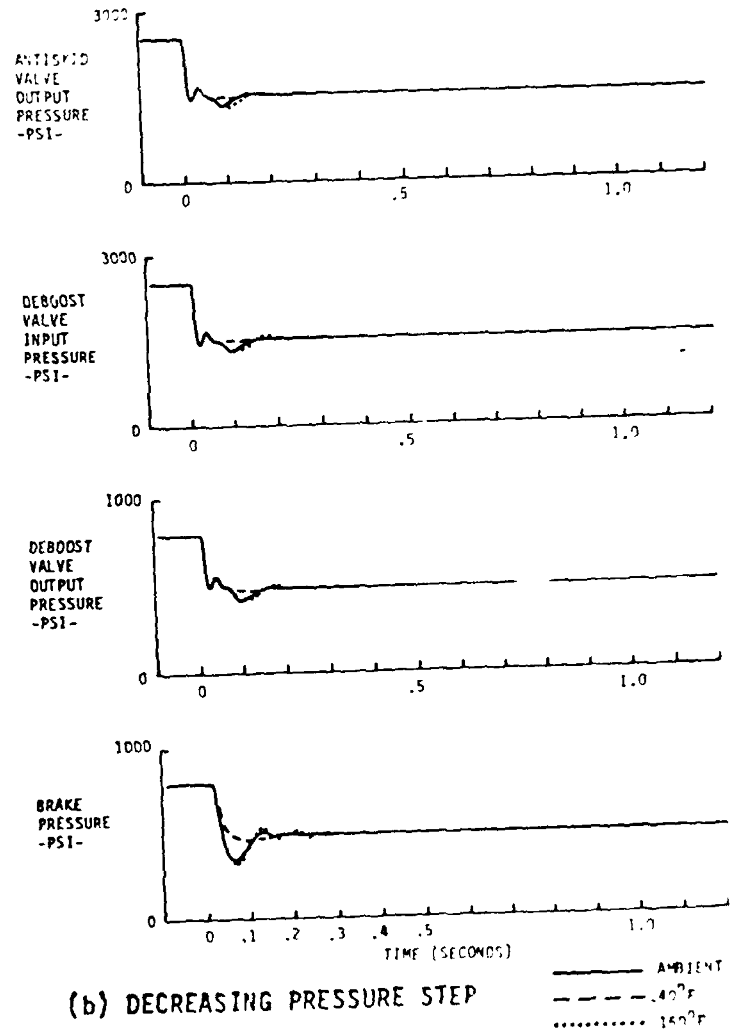


Figure E-62 Step Response, Two-Fluid System, 20-100%



(a) INCREASING PRESSURE STEP



(b) DECREASING PRESSURE STEP

Figure E-63 Step Response, Two-Fluid System, 50-80%

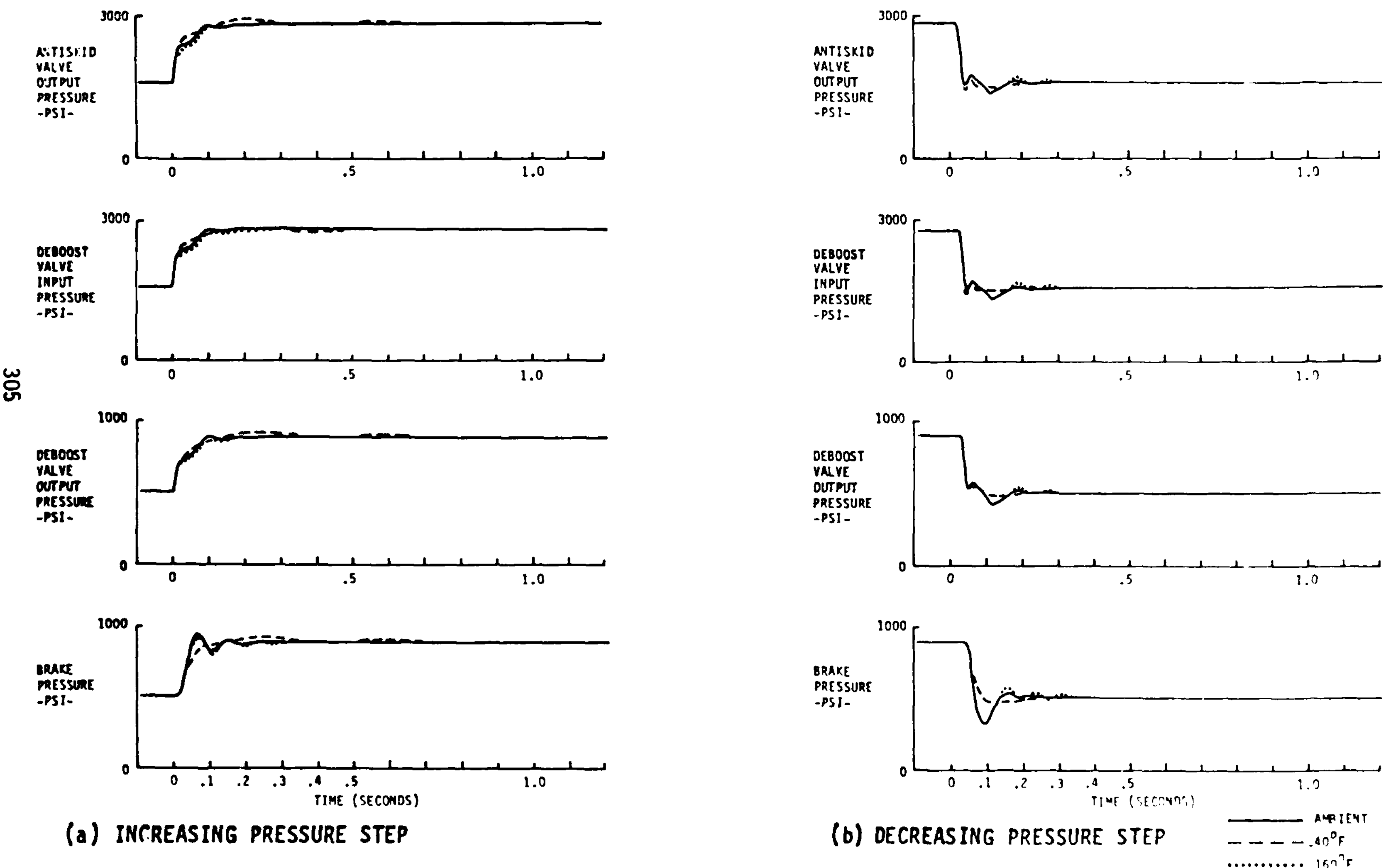


Figure E-64 Step Response, Two-Fluid System, 50-100%

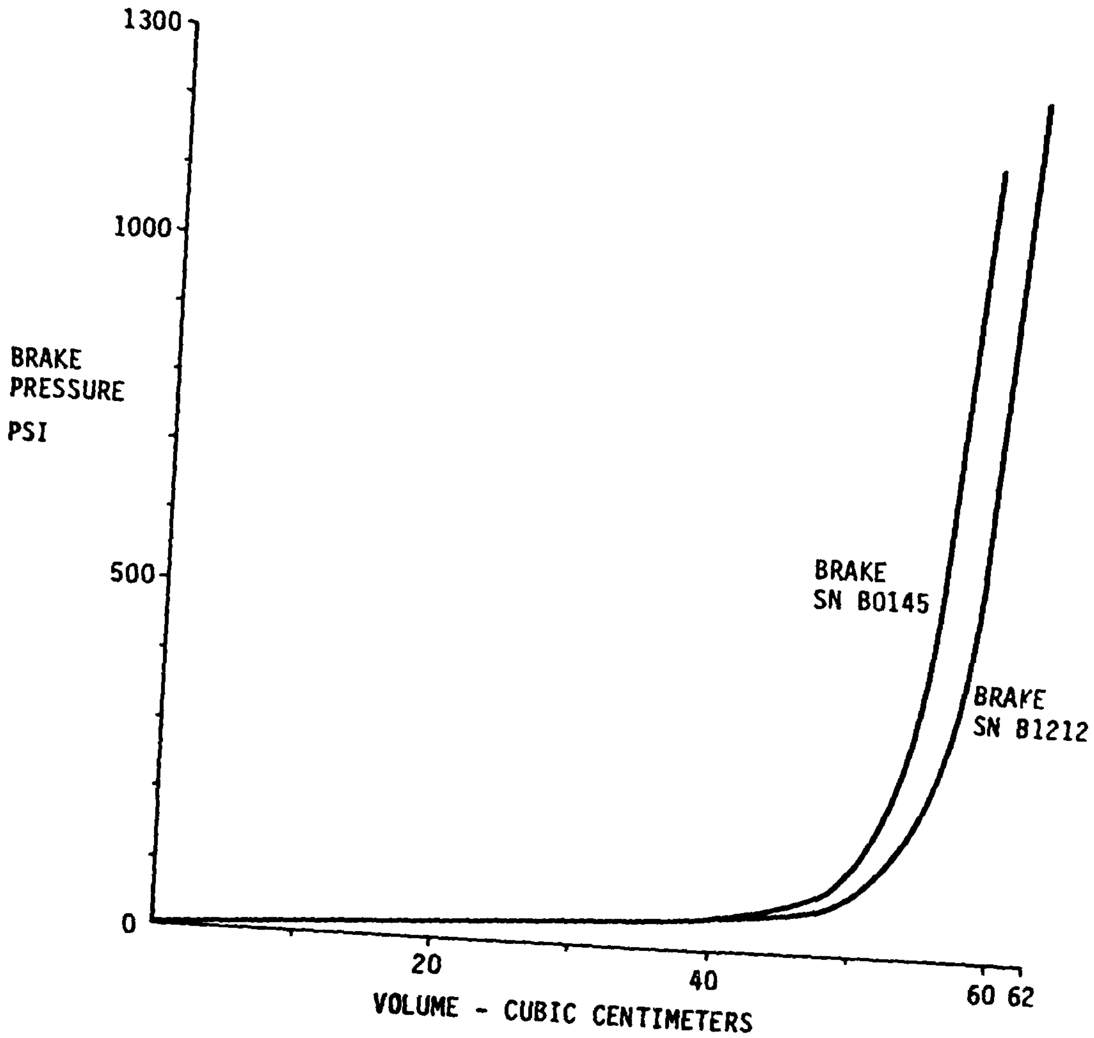


Figure E-65 Brake Pressure Versus Brake Volume, CTFE Fluid Brakes

E.2.5 CONSTANT FRICTION RUNWAY, TEST 5

The stopping performance of the two-fluid brake system was determined as a function of the runway friction coefficient and at ambient, -65 degrees Fahrenheit, -40 degrees Fahrenheit and 160 degrees Fahrenheit. The test results are given in Table E-7. Typical time history plots of wheelspeed, brake pressure and antiskid valve current at constant runway friction coefficients of .5, .3 and .1 and ambient temperature are given in Figures E-66, E-67 and E-68. Similar low temperature (-40 degrees F) and high temperature (+160 degrees F) results are given in Figures E-69 thru E-74.

E.2.6 WET RUNWAY, TEST 6

The two-fluid brake system wet runway test was performed at ambient, -65 degrees Fahrenheit, -40 degree Fahrenheit and 160 degrees Fahrenheit. The stopping distance associated with each test condition is given in Table E-7. Time history plots of wheel speed, brake pressure and the antiskid valve current for the .1 to .5 friction case at ambient, -40 degrees Fahrenheit and +160 degrees Fahrenheit are given in Figures E-75, E-76 and E-77.

E.2.7 STEP FRICTION, TEST 7

The performance and response of the two-fluid brake system to step changes in runway friction was determined at ambient, -65 degrees Fahrenheit, -40 degrees Fahrenheit and 160 degrees Fahrenheit. The stopping distance associated with each test condition is given in Table E-7. Time history plots of wheel speed, brake pressure antiskid valve current and the peak runway friction coefficient at each test temperature are given in Figures E-78, E-79 and E-80.

E.2.8 LANDING GEAR SYSTEM STABILITY, TEST 8

The extent to which the two-fluid brake hydraulic system and antiskid system contributes to the stability of the landing gear strut and brake system was determined. The test was performed at ambient, -65 degrees Fahrenheit, -40 degrees Fahrenheit and +160 degrees Fahrenheit. Results of the test are given

TABLE E-7 STOPPING DISTANCE, TWO-FLUID BRAKE SYSTEM

TEST/DESCRIPTION	FRICTION LEVEL	STOPPING DISTANCE -FEET (BRAKES ON TO 24 fps) AMBIENT 70°F	-65°F	-40°F	+160°F
TEST 5 CONSTANT RUNWAY FRICTION	.6	1935	2489	2066	1901
	.5	2353	2632	2382	2296
	.4	3120	3237	2837	3197
	.3	4816	4118	3604	4668
	.2	8508	5427	5422	7520
	.1	13320	9392	9799	12430
TEST 6 WET RUNWAY - FRICTION AS A FUNCTION OF AIRCRAFT VELOCITY					
	.1 - .5	5543	4614	4380	5028
	.1 - .35	7098	5401	5255	6453
TEST 7 STEP FRICTION	.1 - .5	4521	6694	5917	4133

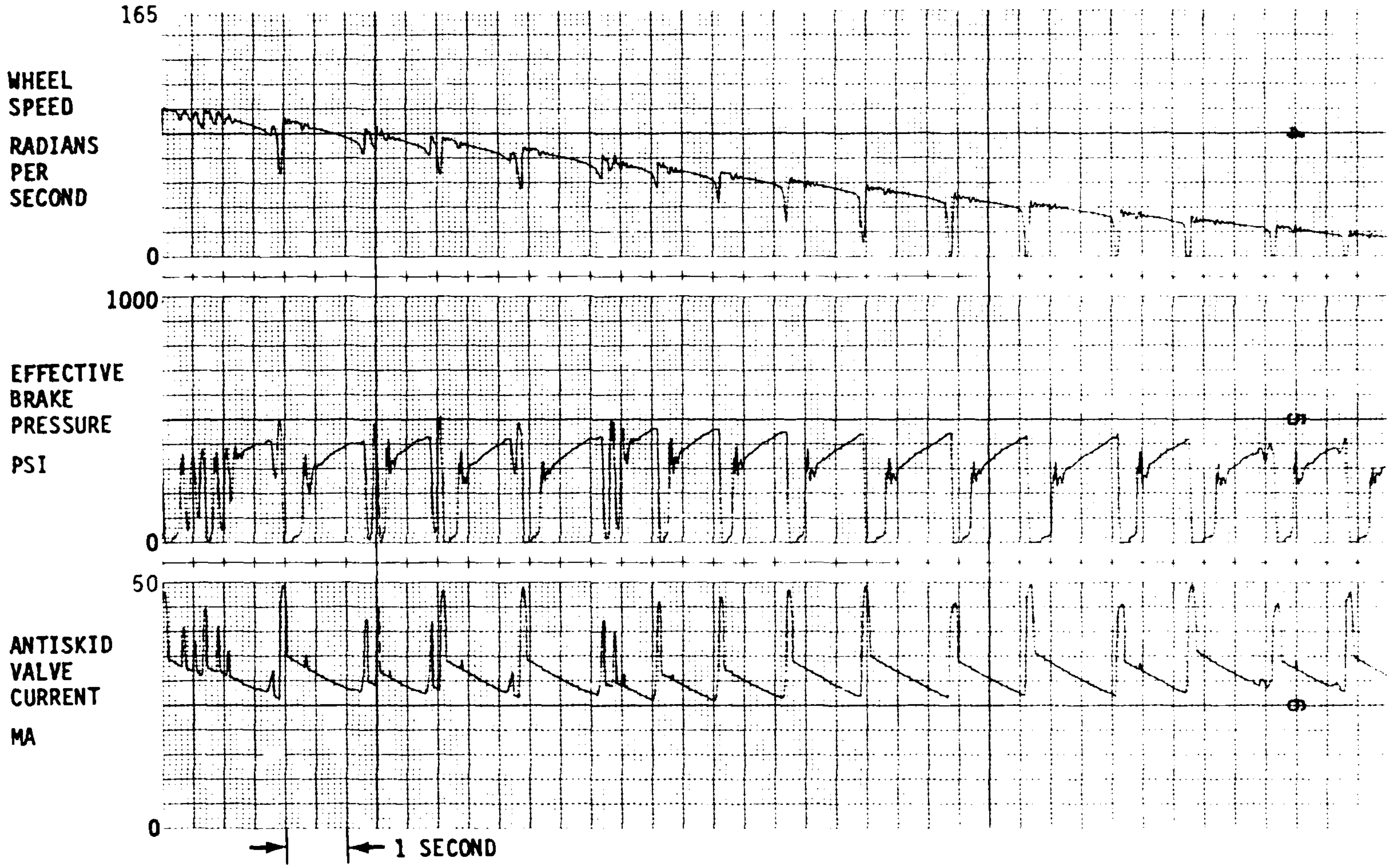


Figure E-66 Brake System Performance, Two-Fluid System, MU=.5, Ambient

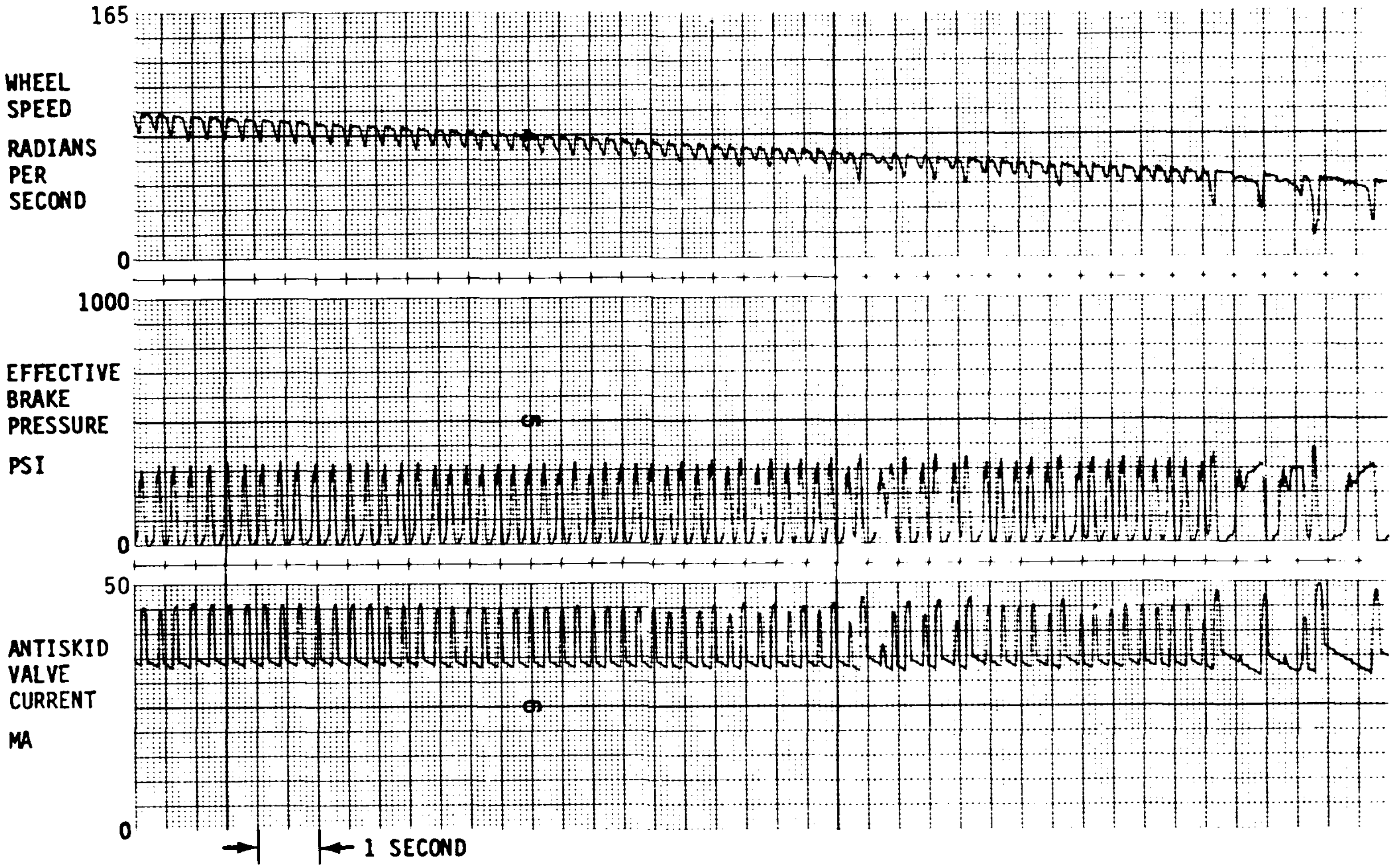


Figure E-67 Brake System Performance, Two-Fluid System, MU=.3, Ambient

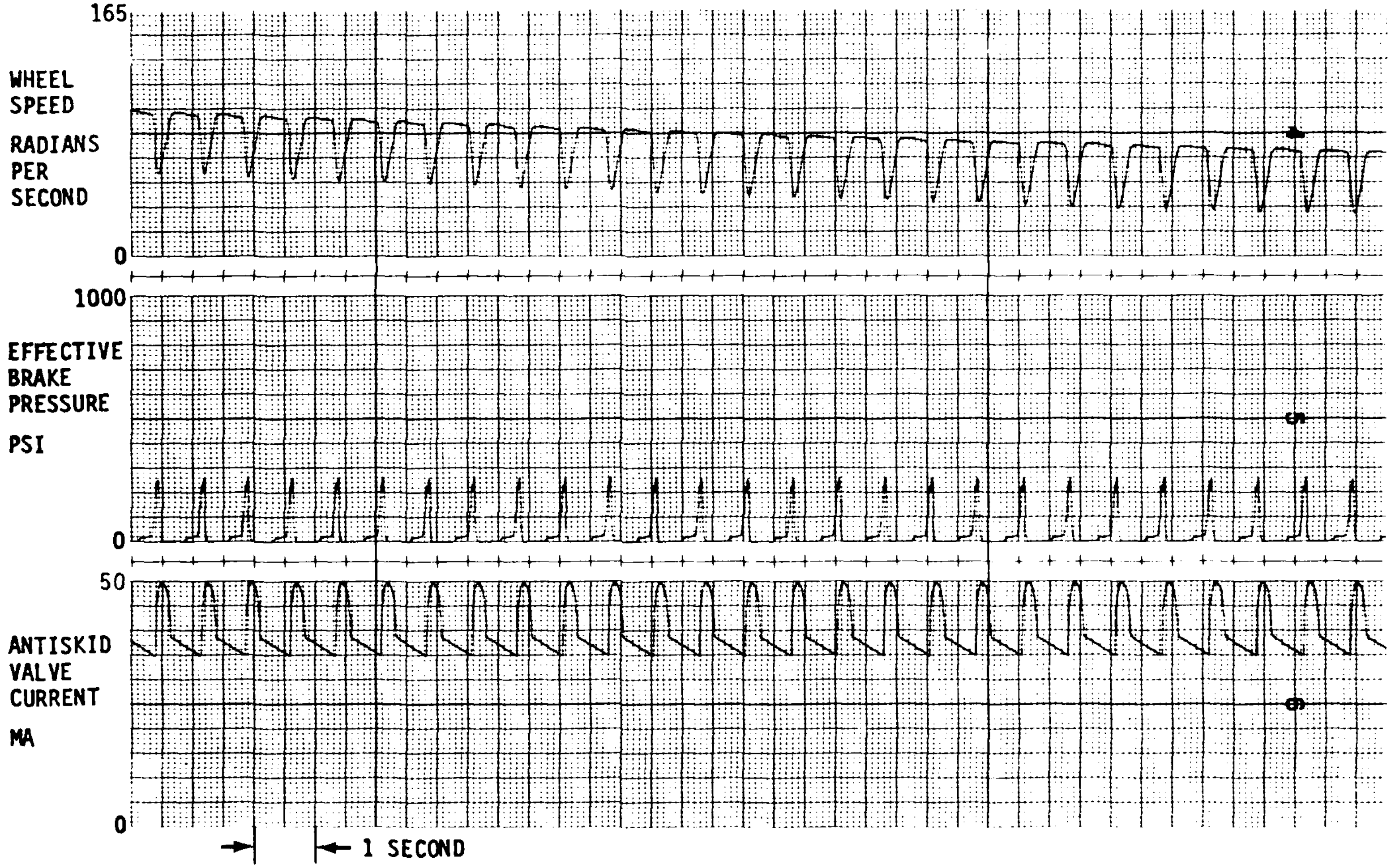


Figure E-68 Brake System Performance, Two-Fluid System, MU=.1, Ambient

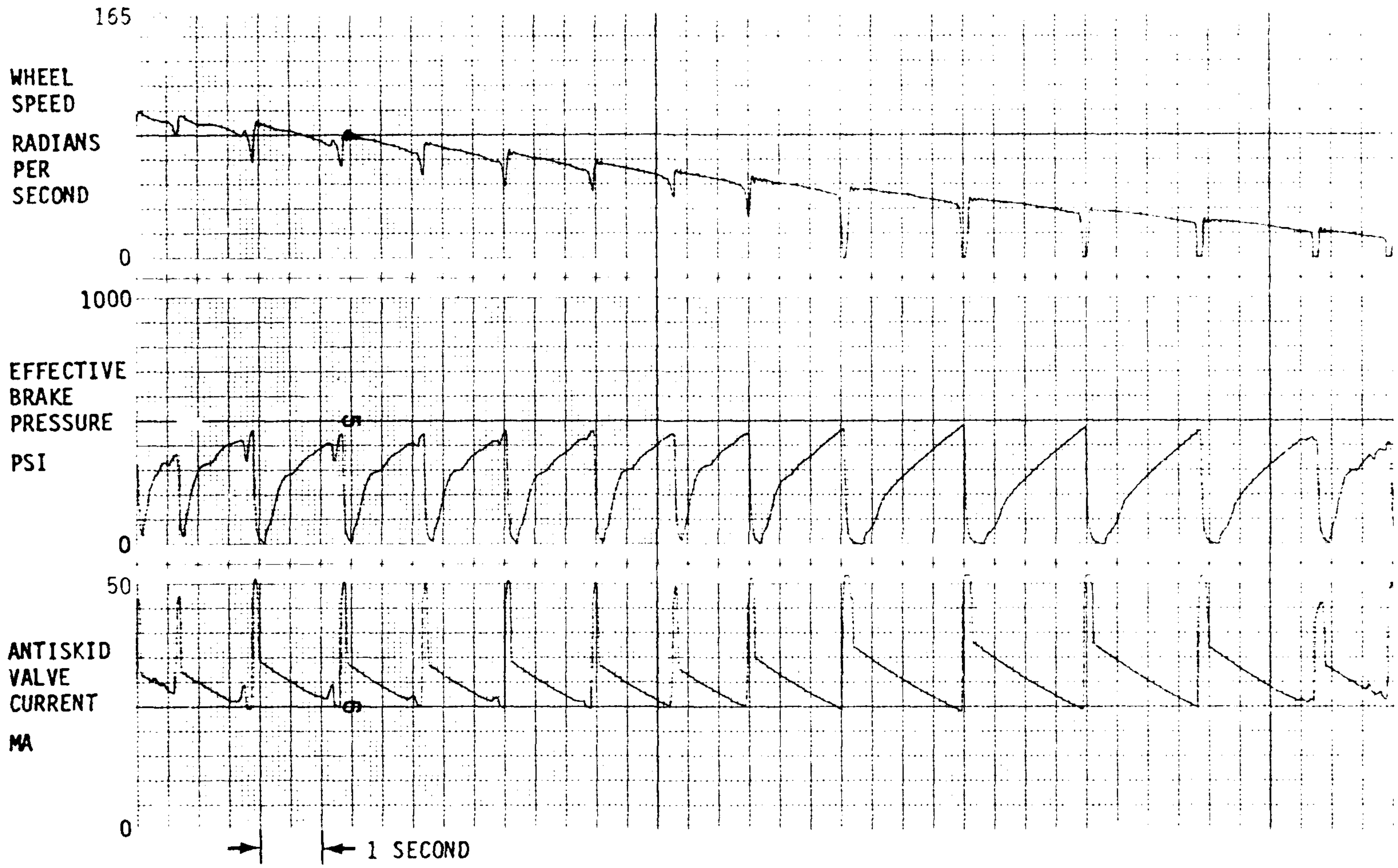


Figure E-69 Brake System Performance, Two-Fluid System, MU=.5, -40°F

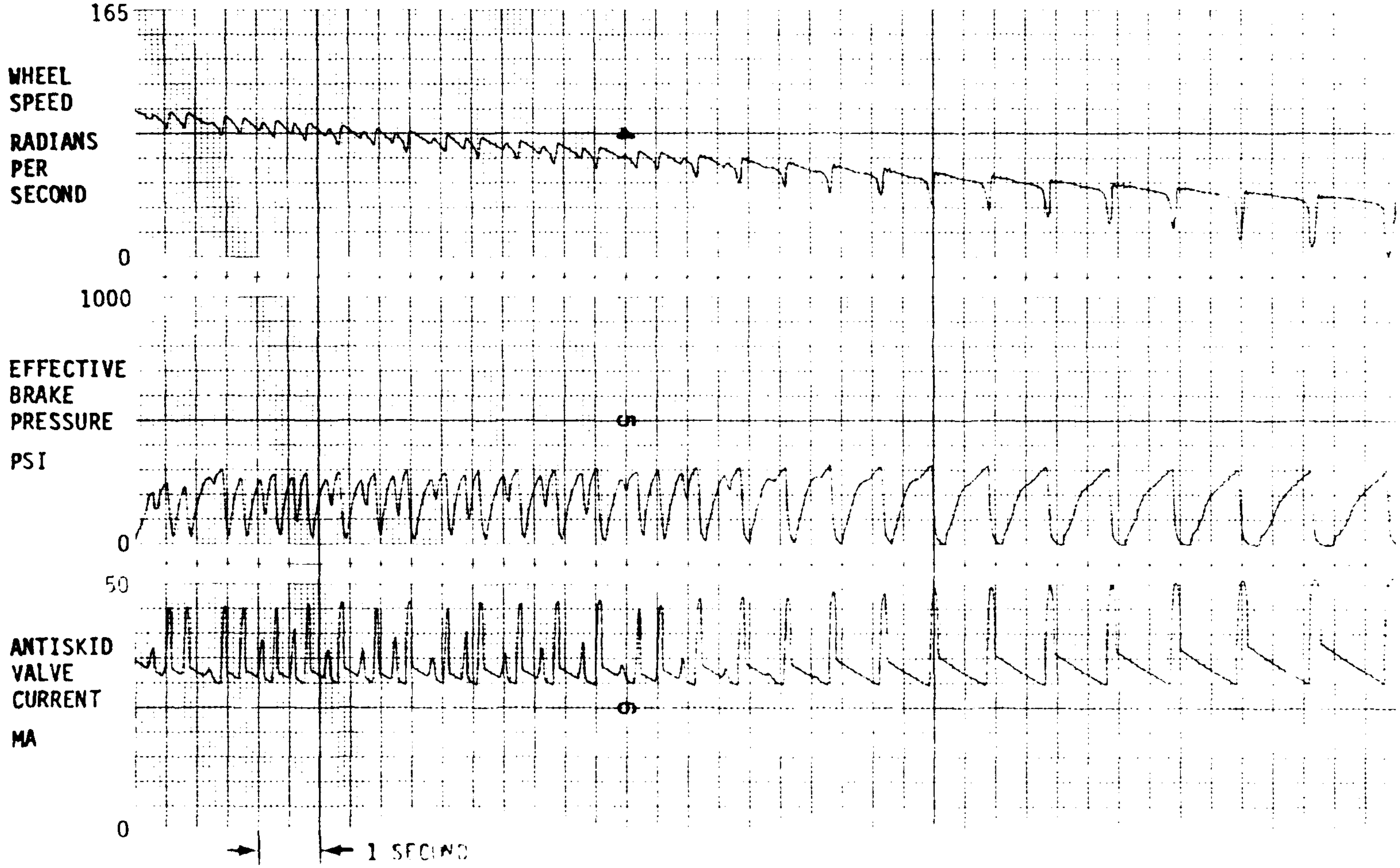


Figure E-76 Brake System Performance, Two-Fluid System, MU=.3, -40°F

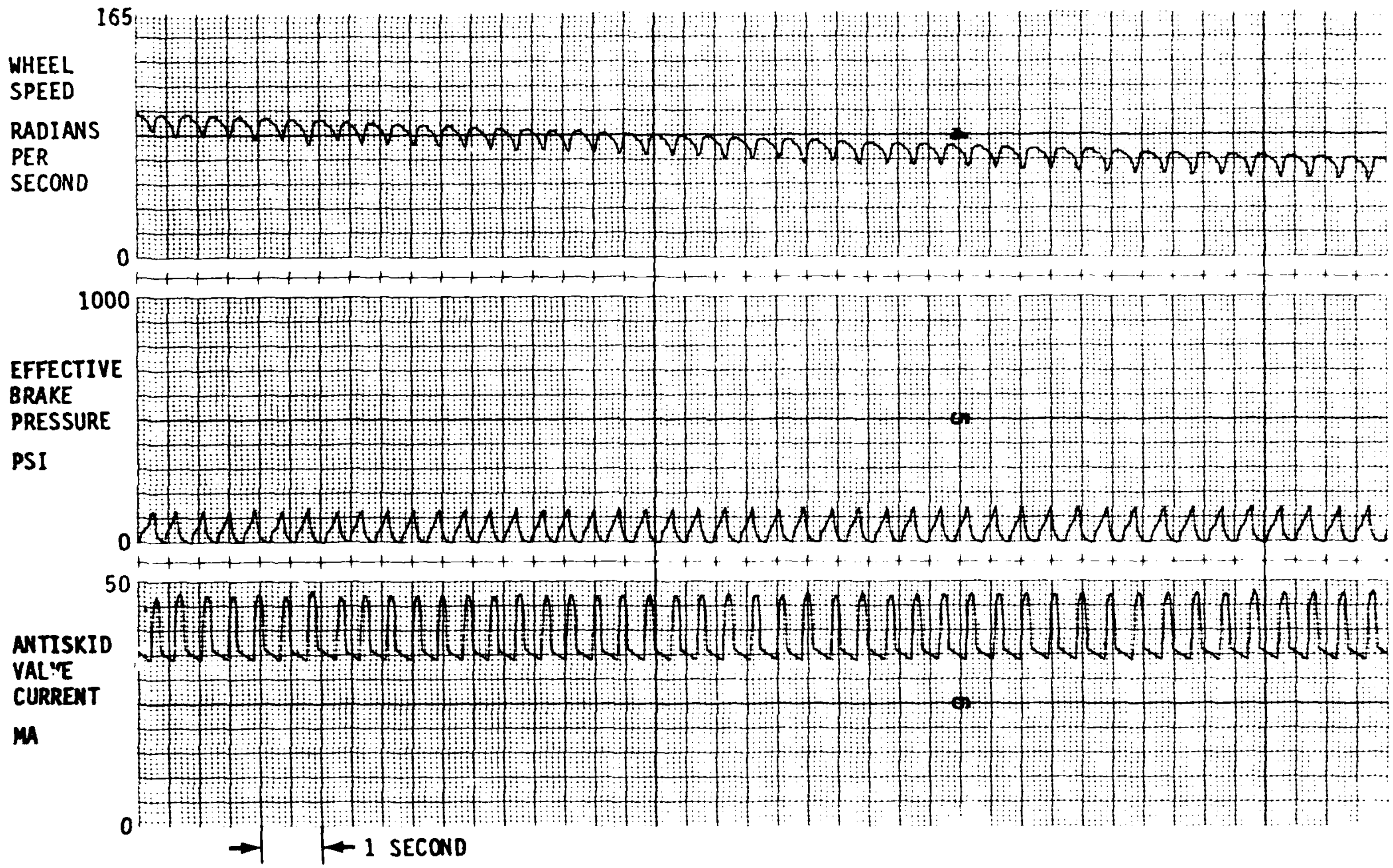


Figure E-71 Brake System Performance, Two-Fluid System, MU=.1, -40°F

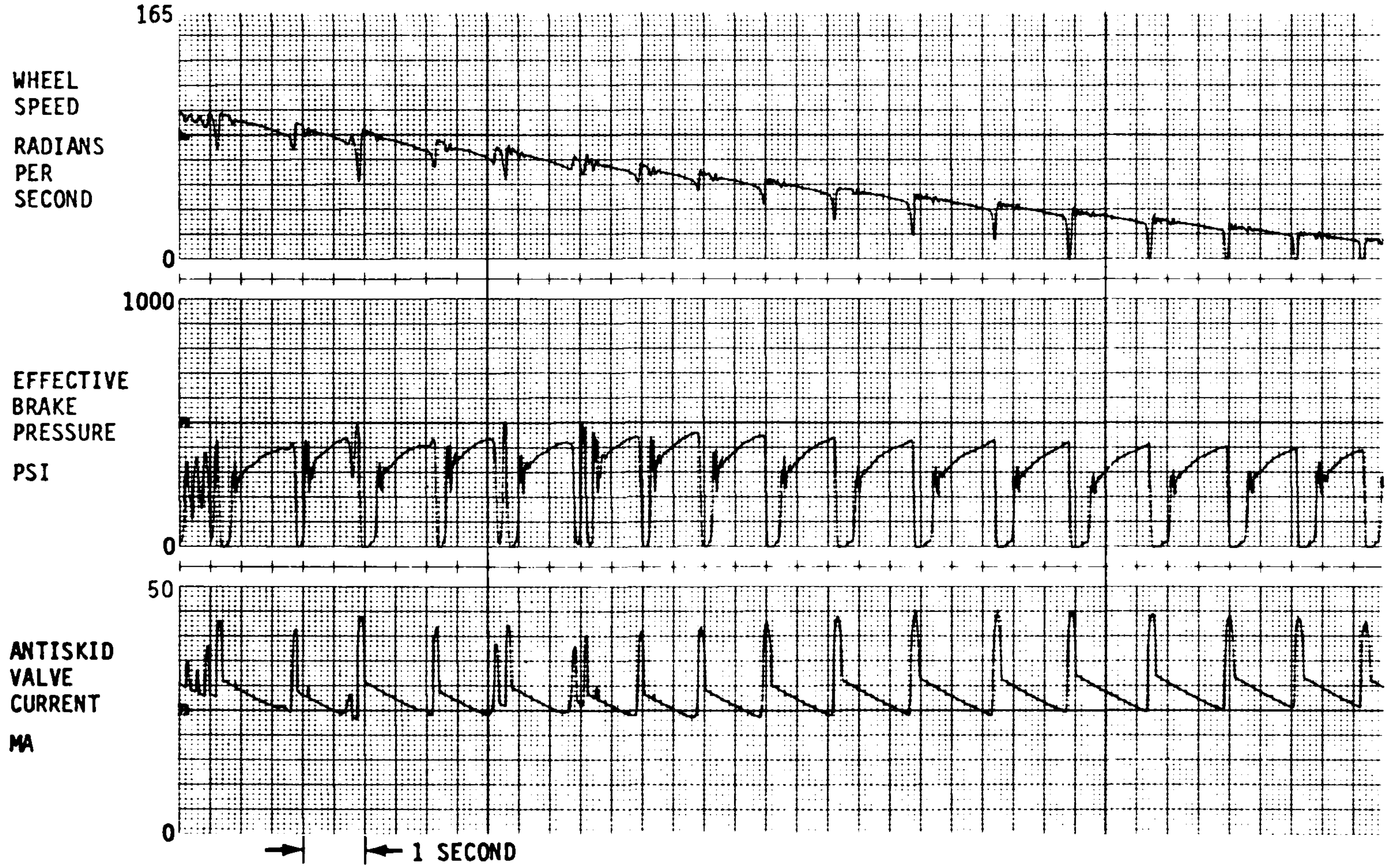


Figure E-72 Brake System Performance, Two-Fluid System, MU=.5, 160°F

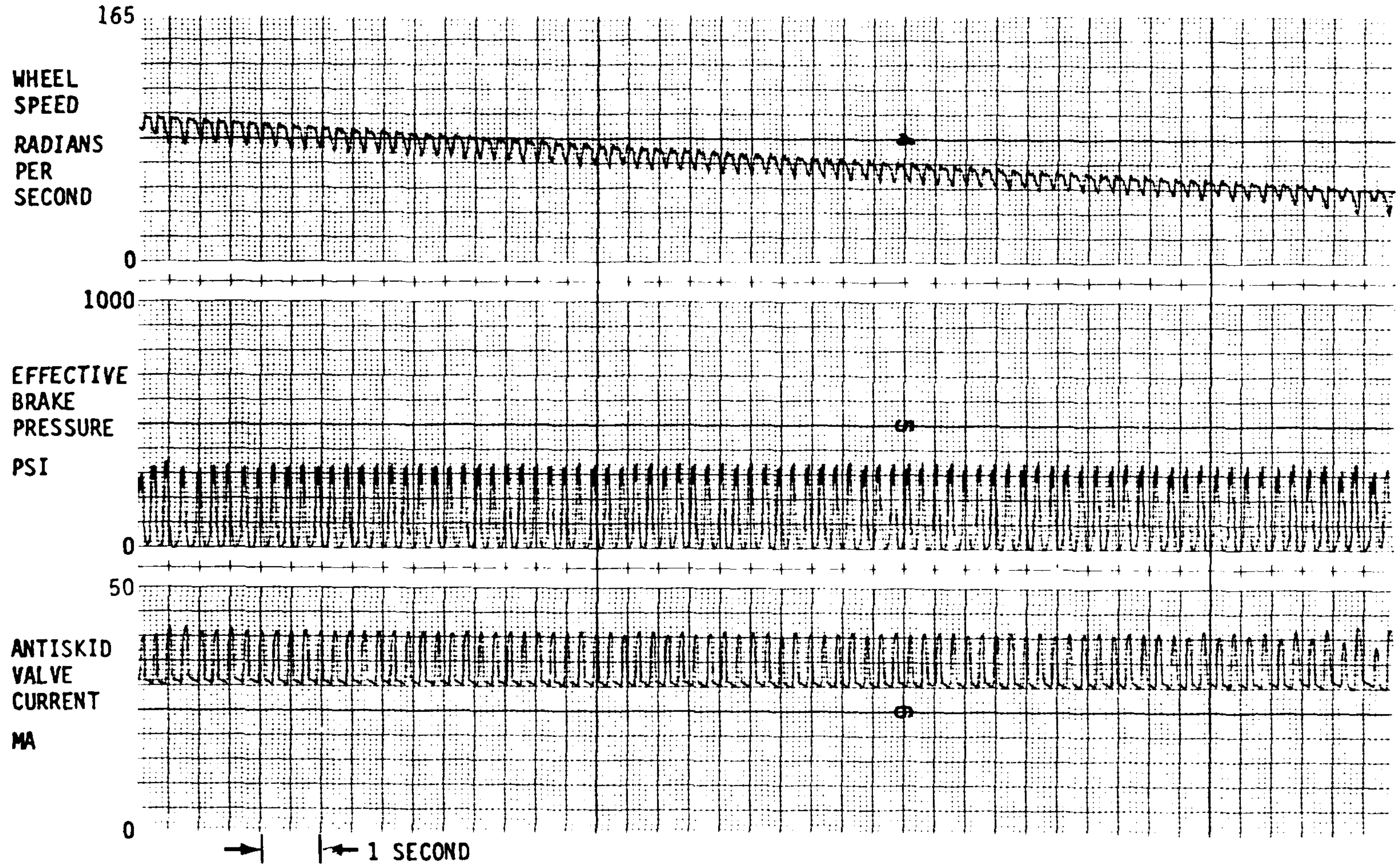


Figure E-73 Brake System Performance, Two-Fluid System, MU=.3, 160°F

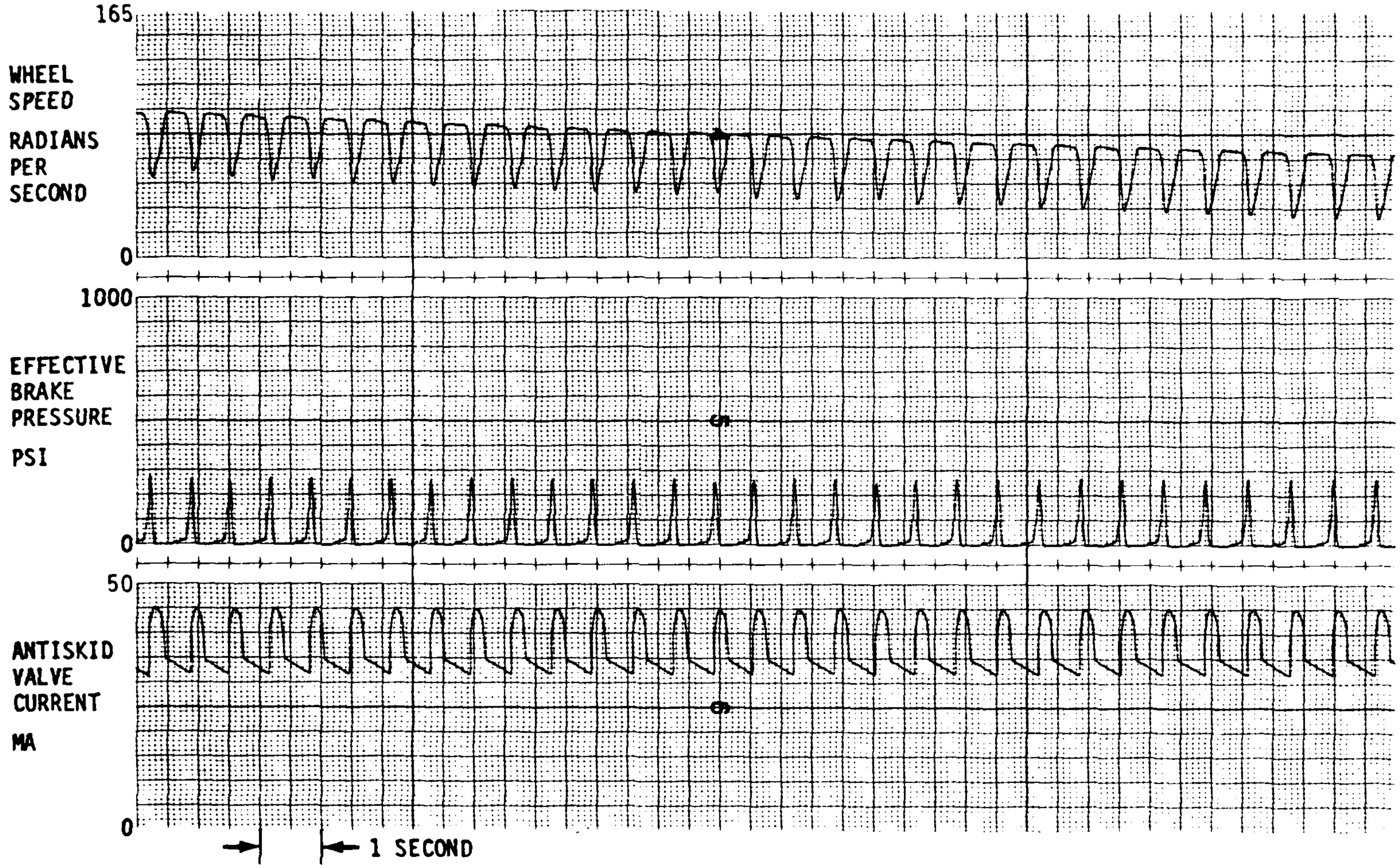


Figure E-74 Brake System Performance, Two-Fluid System, MU=.1, 160°F

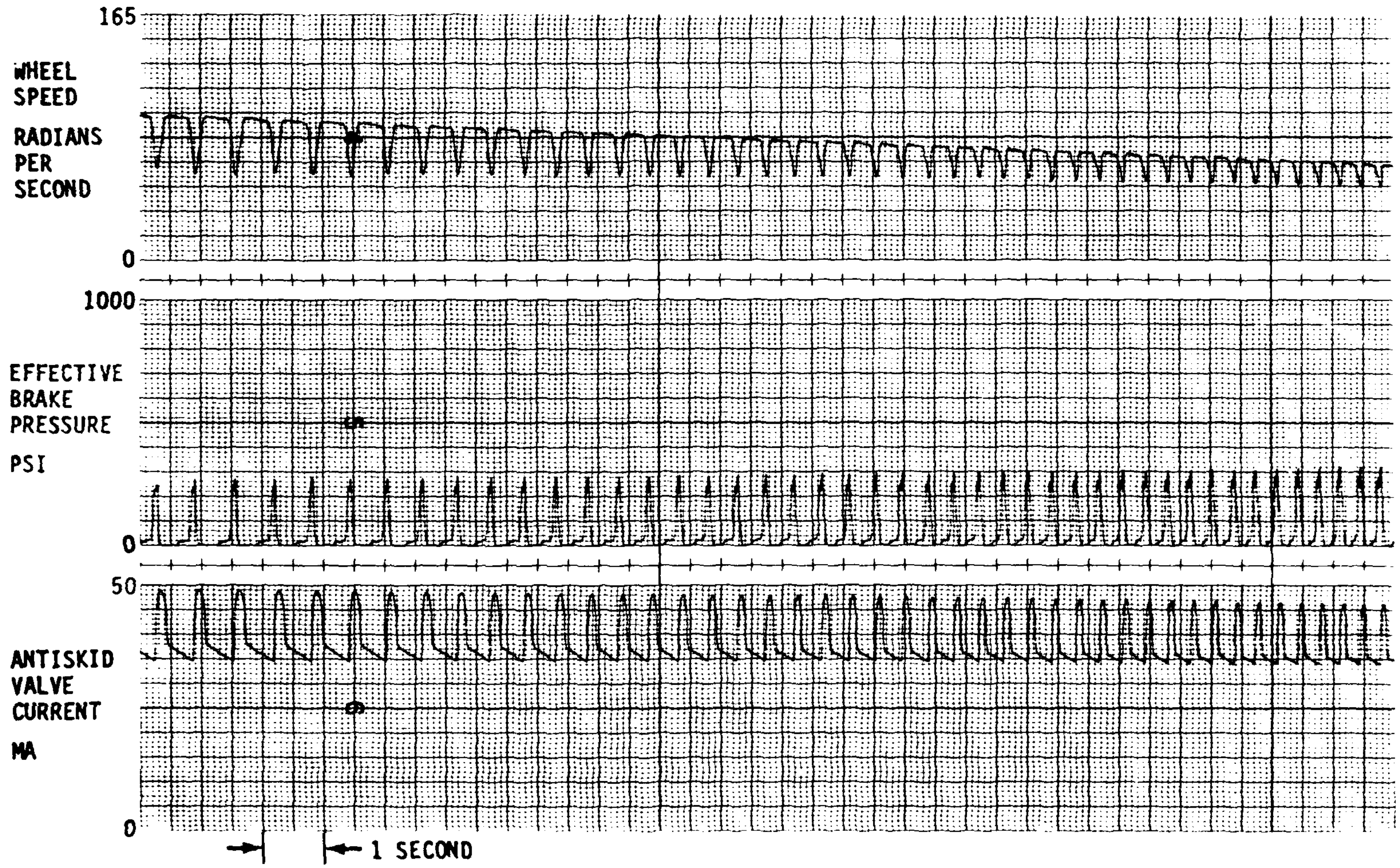


Figure E-75 Wet Runway Performance, Two-Fluid System, MU=.1 to .5, Ambient

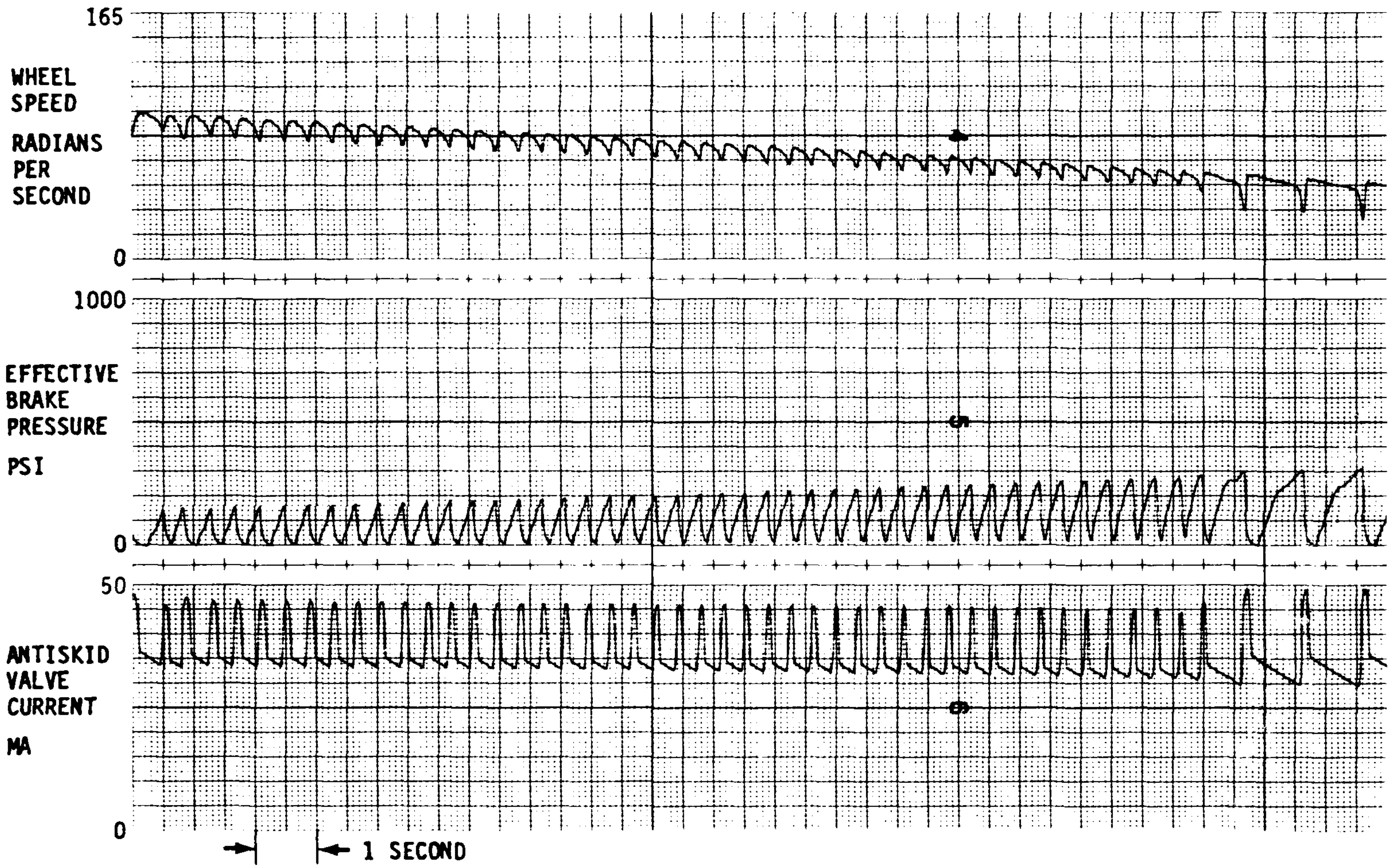


Figure E-76 Wet Runway Performance, Two-Fluid System, MU=.1 to .5, -40°F

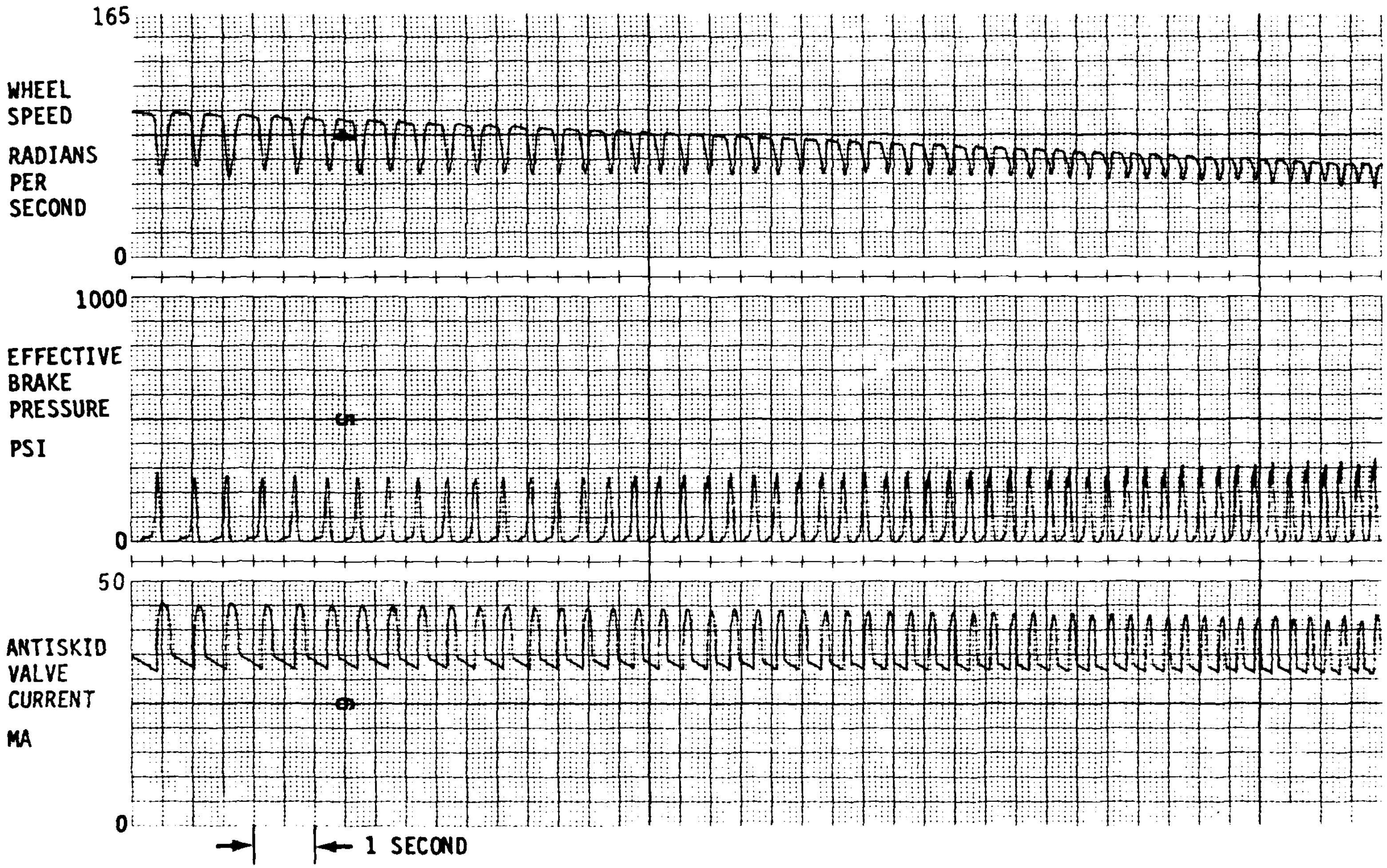


Figure E-77 Wet Runway Performance, Two-Fluid System, MU=.1 to .5, 160°F

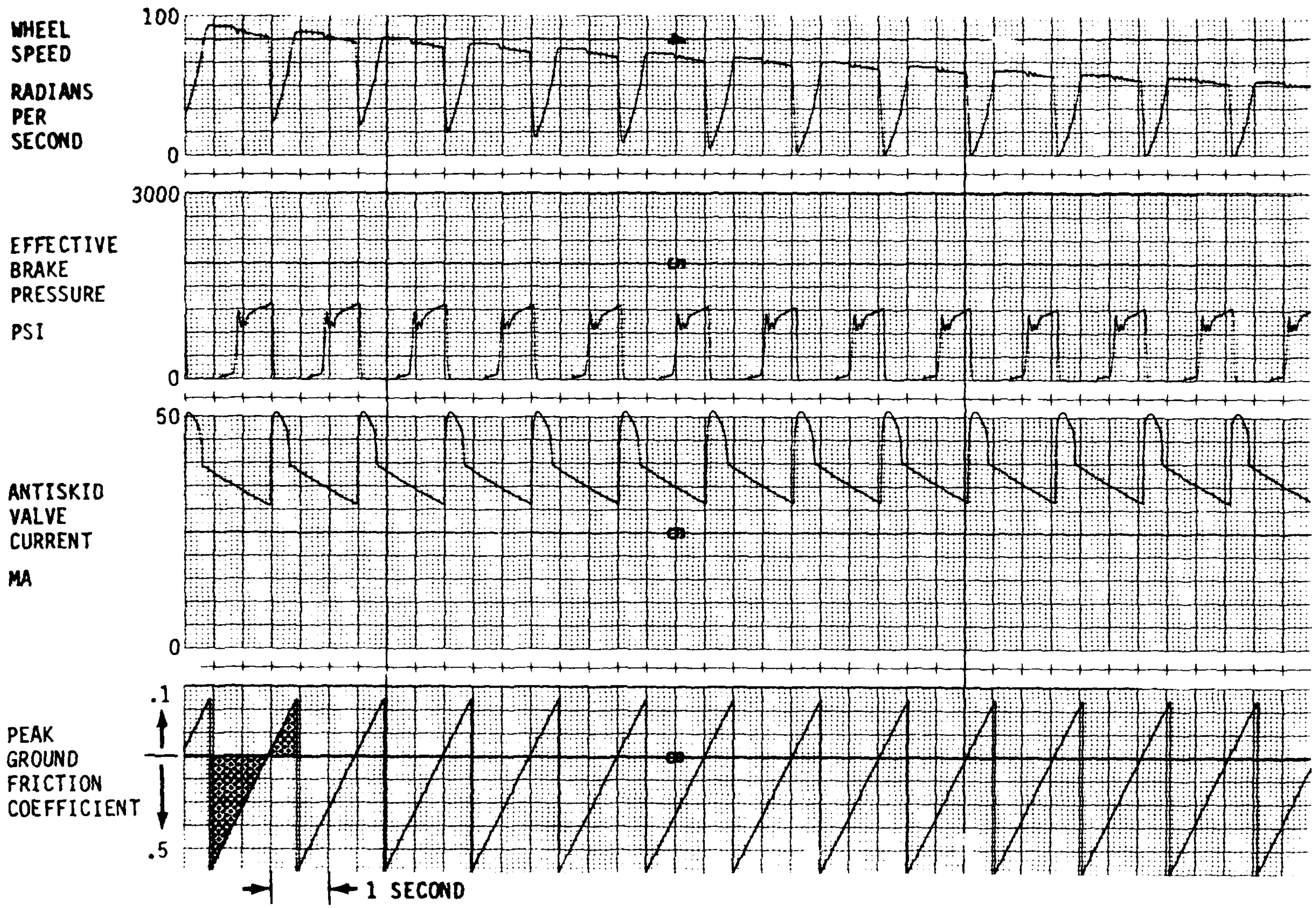


Figure E-78 Step Friction Performance, Two-Fluid System, Ambient

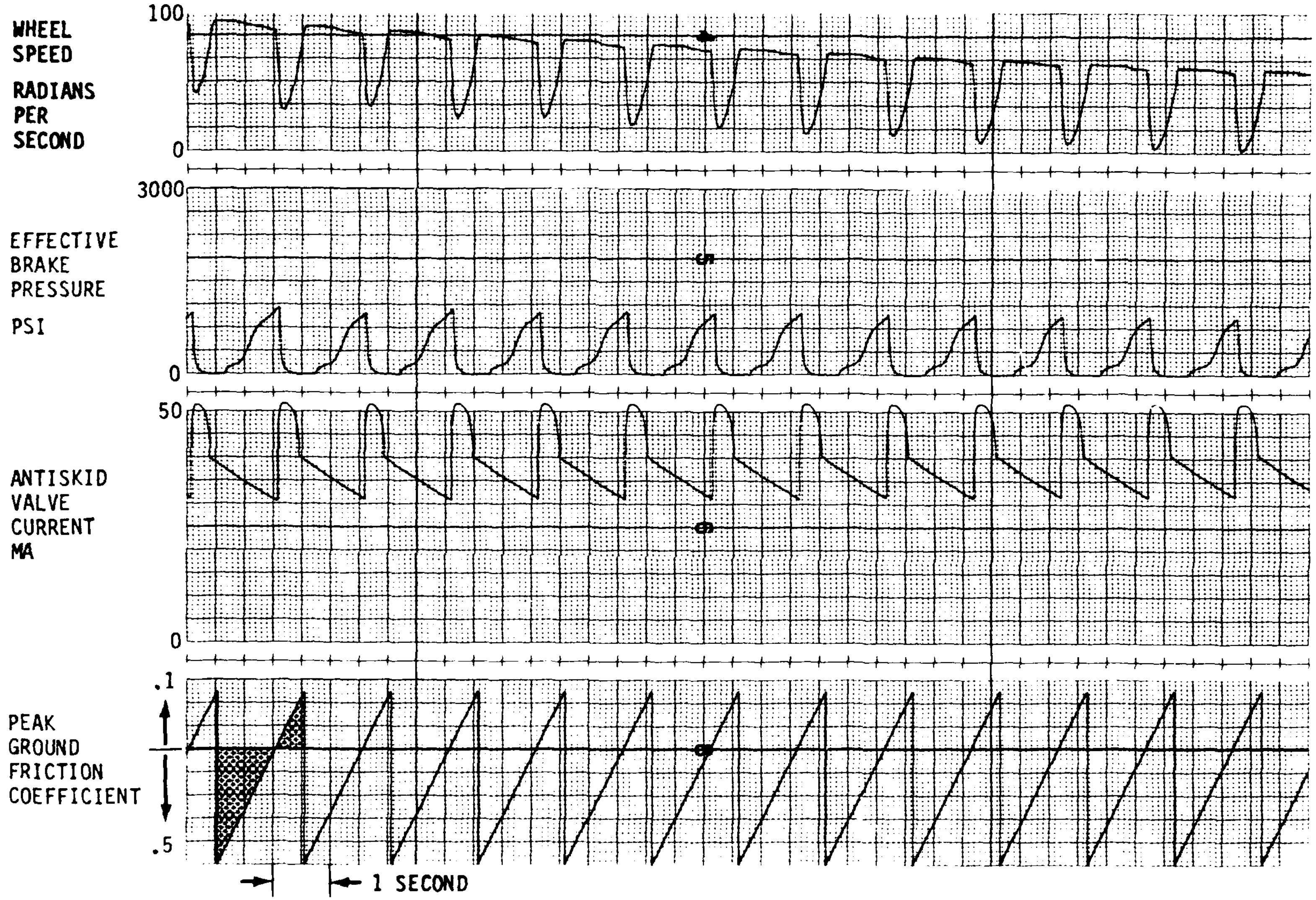


Figure E-79 Step Friction Performance, Two-Fluid System, -40°F

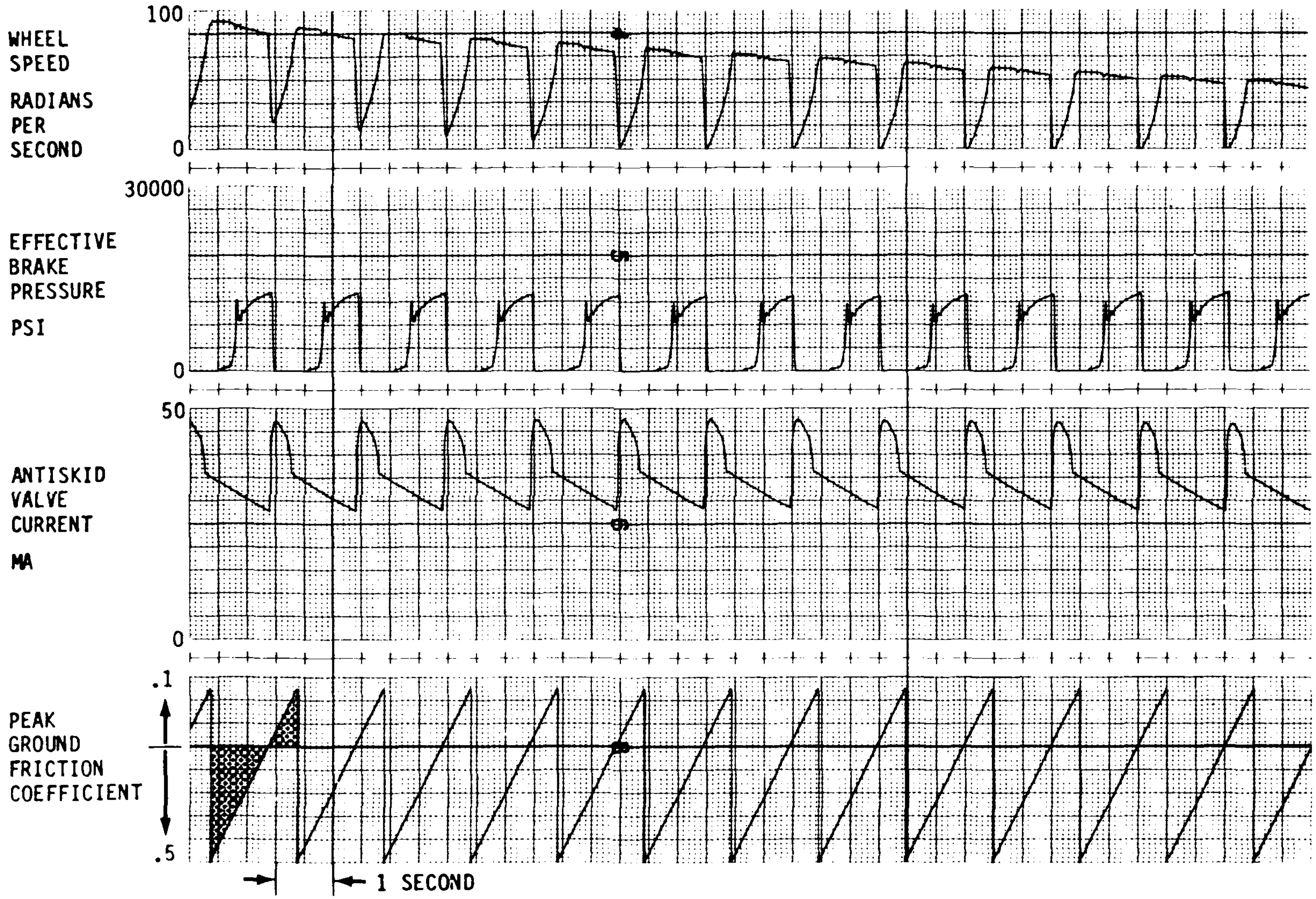


Figure E-80 Step Friction Performance, Two-Fluid System, +160°F

in Table E-8. It was found that the strut damping ratio could be reduced to zero (the normal strut damping ratio is .1) without affecting the stability of the landing gear and brake system. Time history plots of wheel speed, brake pressure and strut displacement at ambient temperature with normal strut damping and with zero strut damping are given in Figures E-81 and E-82.

E.2.9 CTFE FLUID SAMPLES

CTFE fluid samples were taken from the two-fluid brake hydraulic system at regular intervals. These samples were supplied to AFWAL/MLBT for analysis. A complete fluid sample history is given in Appendix H.

TABLE E-8 LANDING GEAR SYSTEM STABILITY, TWO-FLUID SYSTEM

TEST CONDITION		COMMENTS
TEMPERATURE	FORE-AFT DOF STRUT DAMPING RATIO	
AMBIENT	.1 (NORMAL)	Strut oscillations are damped
	0	Strut oscillations are undamped at low speed, strut oscillation amplitude is increased, strut oscillation superimposes ripple on wheel speed, small spikes in antiskid valve current indicates antiskid control box is interpreting strut oscillation as skids
-40	.1 (NORMAL)	Strut oscillations are damped
	0	Strut oscillations are undamped at low speed, strut oscillation superimposes ripple on wheel speed, strut oscillations are interpreted as skids
160	.1 (NORMAL)	Strut oscillations are damped
	.01	Strut oscillations are undamped at low speed, strut oscillation superimposes ripple on wheel speed, strut oscillations are interpreted as skids

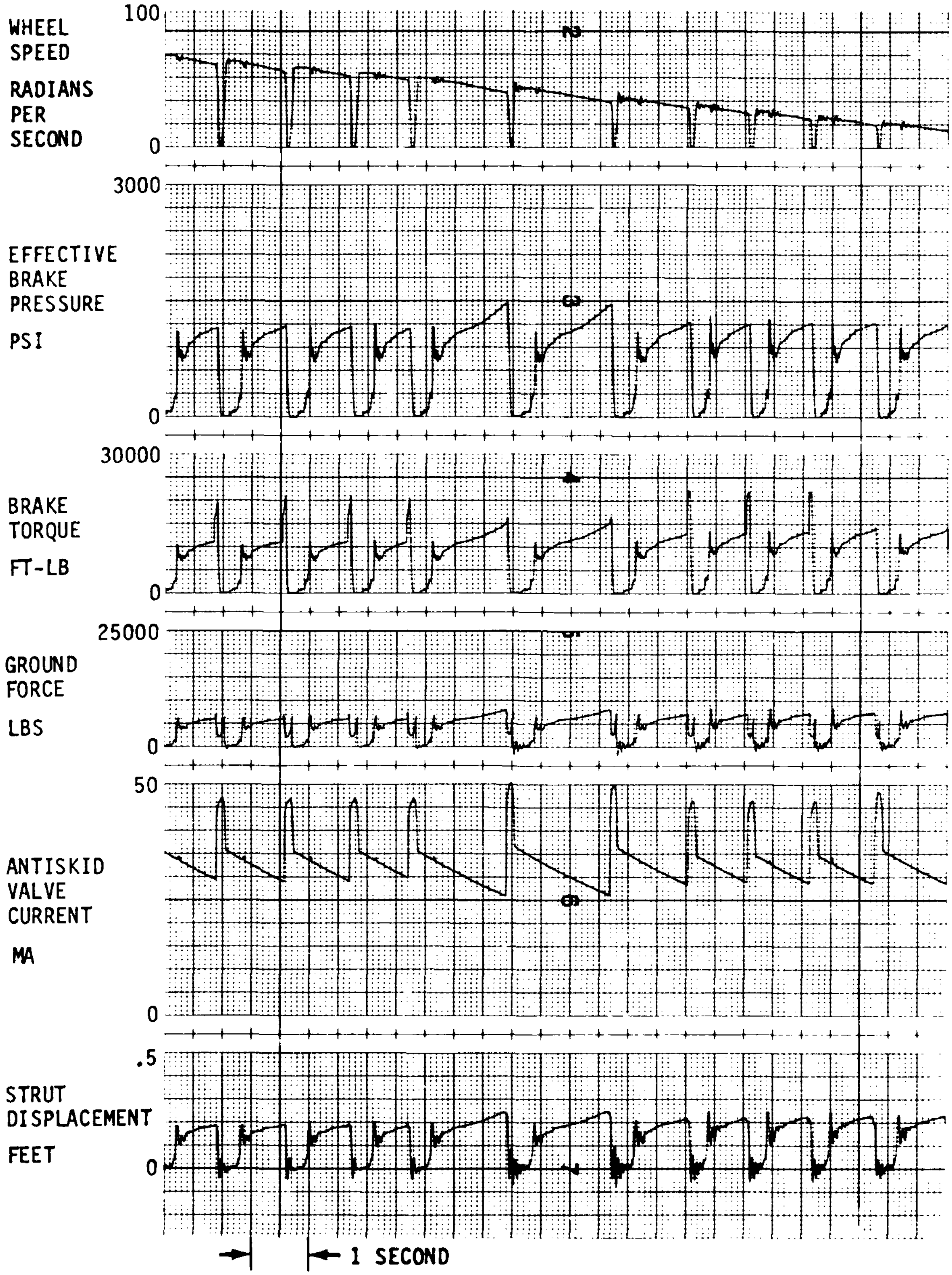


Figure E-81 Brake System Stability, Two-Fluid System, Normal Damping Ambient

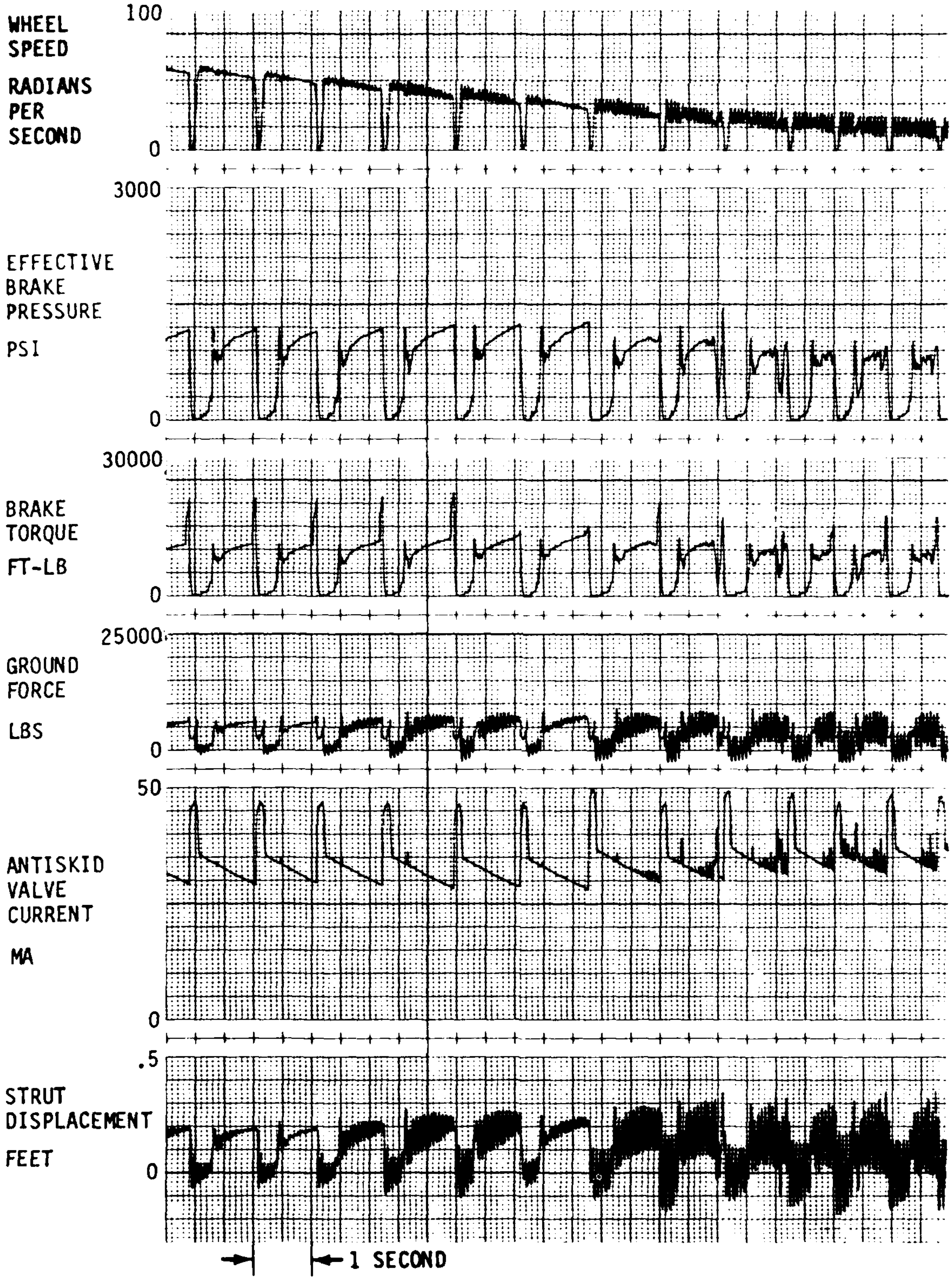


Figure E-82 Brake System Stability, Two-Fluid System Zero Damping, Ambient

APPENDIX F

HYBRID BRAKE CONTROL LABORATORY

The two-fluid brake hydraulic system and the standard brake system were tested in the Boeing Hybrid Brake Control Laboratory (HYBCOL) located at the Boeing Development Center, Seattle, Washington. The primary functions of this facility are to develop new landing gear and brake control system concepts, and checkout, tune, and predict the performance of existing braking systems.

The laboratory is a triple hybrid facility incorporating analog and digital computers and actual aircraft hardware. The HYBCOL enables the user to simulate in real time the response of an aircraft and its brake control system. The digital and analog computers contain mathematical models of aircraft rigid body dynamics and landing gear systems which interface with the hydraulic brake system mockup and antiskid control system that form the hardware elements within the facility. An overall schematic of the laboratory is shown in Figure F-1.

The HYBCOL is presently supported by two Denelcor Model CI-450 Analog/Hybrid Computers, a Data General Eclipse digital minicomputer, a Data General Nova 3 minicomputer, analog-to-digital and digital-to-analog converters, a CRT, line printer and other peripheral equipment. Figure F-2 shows the relationships and communication links between the elements within the simulator. The airplane simulation is divided between the analog and digital computers, with all high frequency components modeled on the analog and low frequency components modeled on the digital computer. This division increases the operational efficiency of the Brake Control Laboratory due to the increased computing efficiency and flexibility which can be achieved. Figure F-3 shows a pictorial view of the digital minicomputer system while Figure F-4 shows the analog computer.

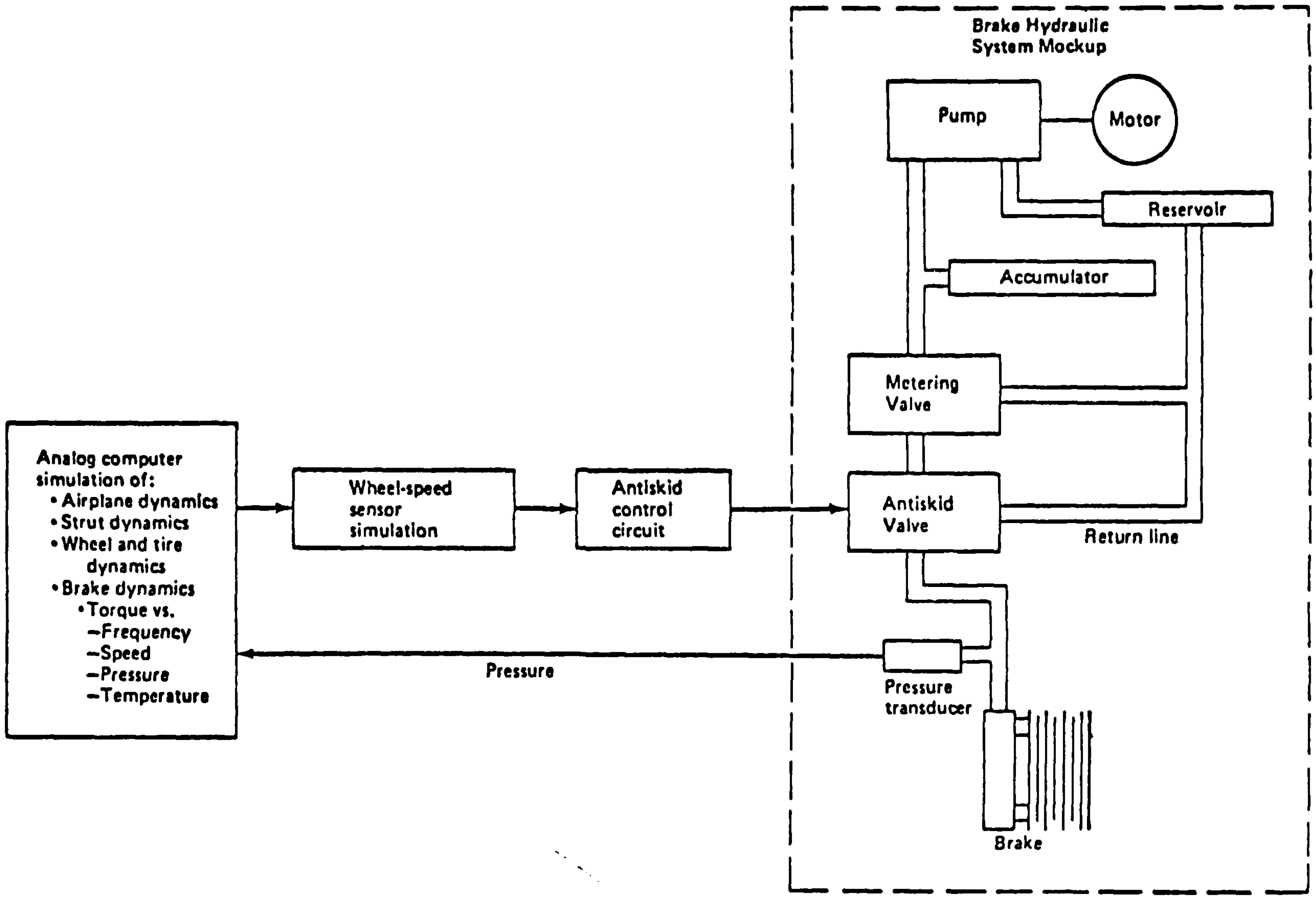


Figure F-1 Antiskid Laboratory Facility - Schematic

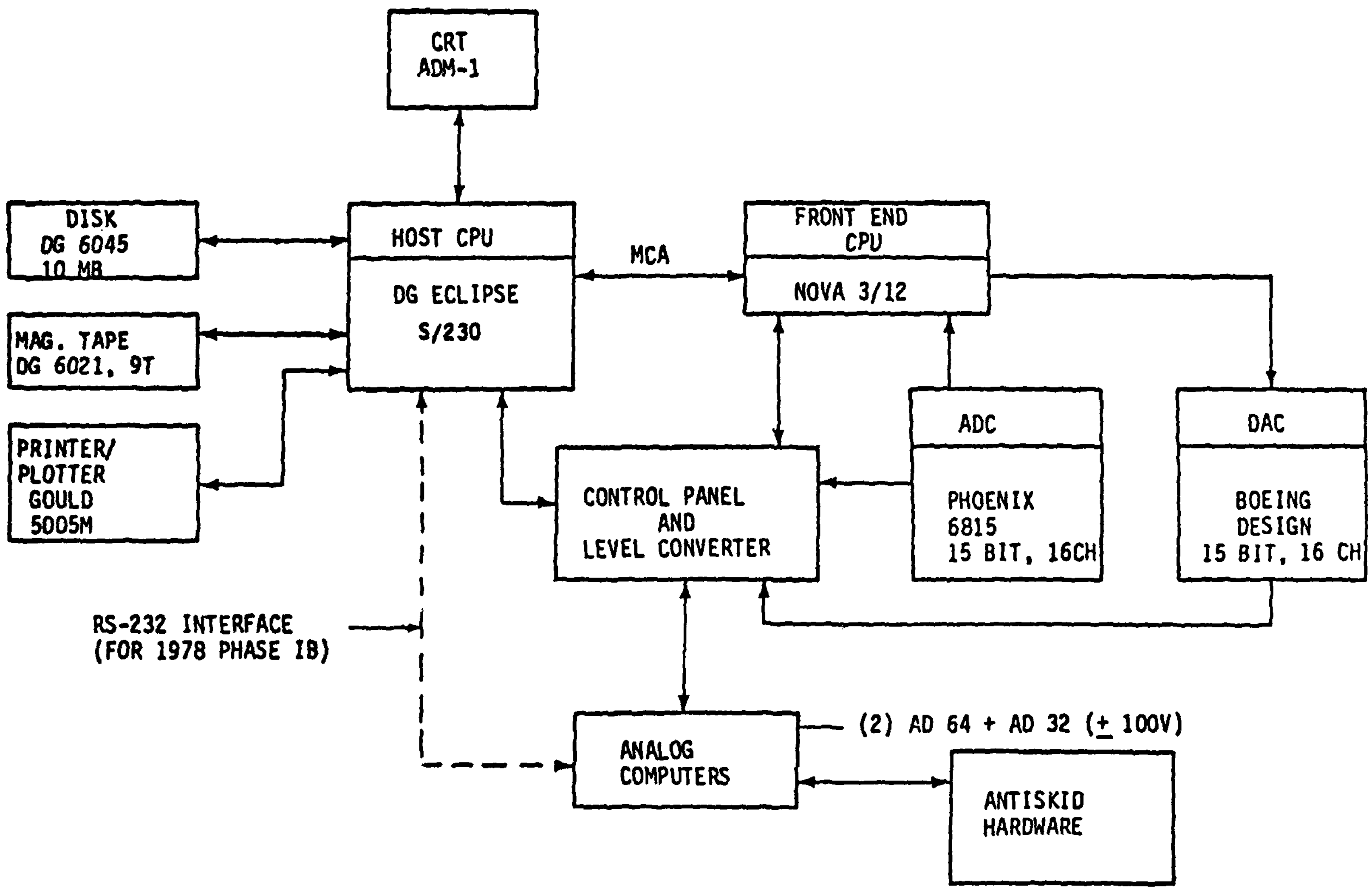


Figure F-2 Antiskid Lab Minicomputer System

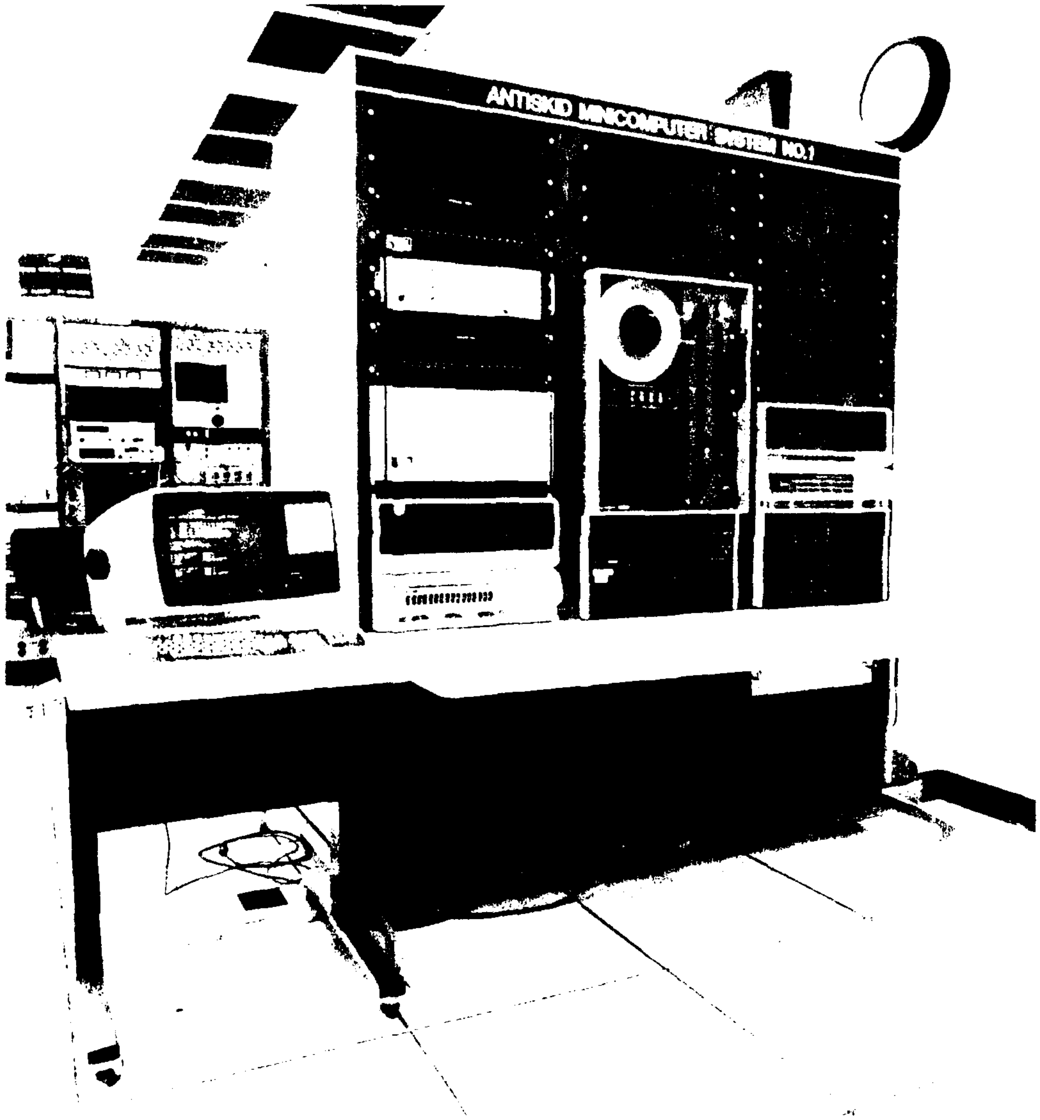


Figure F-3 Digital Minicomputer System.
Hybrid Brake Control Laboratory

Figure F-4 Analog Computers, Hybrid Brake Control Laboratory

On a given airplane program, the laboratory is used to configure, refine, and finally test the performance of an antiskid brake control system. The system characteristics are determined based on:

- o Stopping distance performance
- o Gear and truck stability
- o Adaptability to various runway conditions, and
- o Brake hydraulic response

Based on simulator and initial flight test results, the antiskid control box is tuned (critical control system components, resistors, capacitors, etc. are carefully modified) to provide improved performance, and flight tested again. This process of antiskid tuning and flight testing of the tuned box may go through anywhere from five to ten iterations until an optimum operational configuration is achieved. The simulation support is further carried on through the final FAA certification of the brake control system on the airplane.

In addition to antiskid tuning and certification, the laboratroy facility has been extensively used for the development and certifications of autobrake systems, NASA/Air Force studies on brake control systems, etc. It has also been used for a non-airline application such as the evaluation of Vertol Rail Car antiskid system.

To make maximum use of the facility, the simulation tool best suited to the requirements of each model is used. A typical simulation model structure is listed below:

Digital Minicomputer

- o Airplane rigid body equations of motion
- o Aerodynamics
- o Engines
- o Flight controls

Analog Computers

- o Tire/wheel dynamics
- o Strut dynamics
- o Tire/ground force
- o Brake torque generation

Hardware

- o Hydraulic brake system
- o Antiskid system
- o Miscellaneous data acquisition and signal processing equipment

Typically, the digital computer contains the low frequency calculations associated with the aircraft rigid body equations of motion, aerodynamics, engines, and flight controls. These models generally have frequency content less than 10 Hertz. Strut, wheel brake, and ground force models are typically found on the analog computer. These models are characteristic of higher natural frequency (10 Hertz and above) and cannot be implemented accurately on the digital computer. The hardware which is used in the simulation serve two purposes. The transients and non-linearities associated with the hydraulic system have a significant effect on the performance of the antiskid system. Due to this complexity and importance the brake hydraulic system is mocked-up using actual aircraft hardware. In addition to the hydraulic mock-up, brake-antiskid electronic control hardware is also employed. Use of the mockup and antiskid control box assures that the braking performance of aircraft is accurately reproduced.

This breakdown of the aircraft and landing gear systems model described above best matches the dynamic capabilities of the computer tool with those required by the model, thereby providing efficient use of the analog and digital computers. The advantage of such an approach are many, including:

- o The use of a detailed aircraft simulation along with actual antiskid and hydraulic brake system hardware permits the accurate prediction of the aircraft braking performance.

- Real time simulation allows reduced flow time, allows a large increase in the conditions evaluated and assures a timely completion of the braking performance evaluation.

- Hands-on capability allows the engineer to evaluate the effects of the hydraulic system changes immediately.

- The hybrid facility is a dedicated brake control system laboratory used solely for antiskid and landing gear research, program and contract support.

APPENDIX G

KC-135 COMPUTER SIMULATION

A real time KC-135 aircraft braking simulation was created for the laboratory testing of the two-fluid brake system. The simulation was implemented in the *Boeing Hybrid Brake Control* Laboratory (HYBCOL). A description of the laboratory facility is included as Appendix F.

The KC-135 simulation was developed exclusively for the evaluation of the two-fluid brake hydraulic system. The simulation contains mathematical models of:

- o Three degrees of freedom rigid body vehicle dynamics (longitudinal dynamics; forward, vertical and pitch)
- o Longitudinal aerodynamics
- o Landing gear
- o Ground force
- o Brake torque and
- o Wheel dynamics.

In addition to these mathematical descriptions, the simulation included a mock-up of KC-135 brake hydraulic system hardware and a KC-135 Mark II antiskid control box. A block diagram of the KC-135 simulation is shown in Figure G-1. The major paths of interaction between the models and hardware are shown.

The models and equations utilized in the simulation were developed during previous Air Force contracts and Boeing research efforts. A brief explanation of the models, assumptions and simplifications follows.

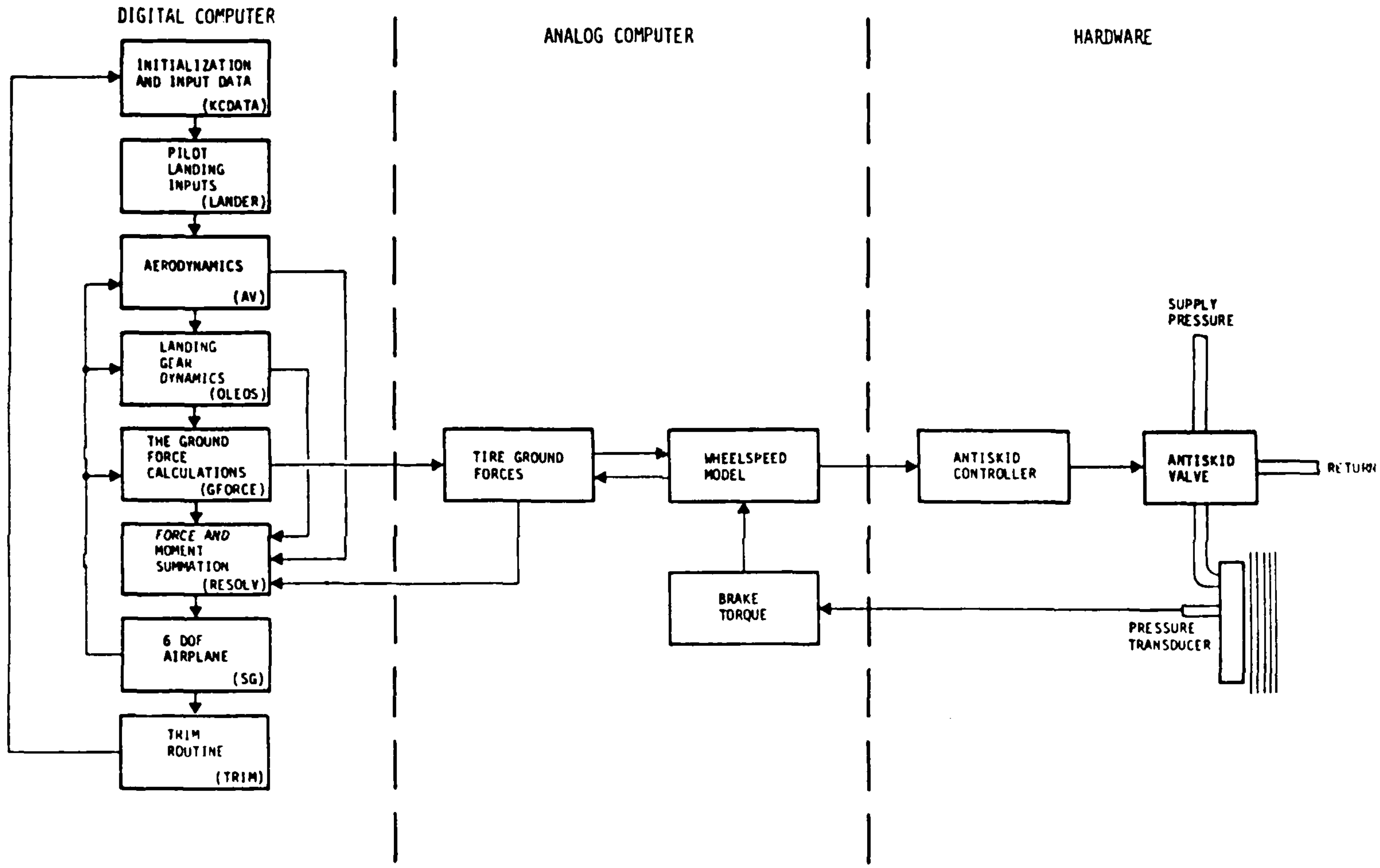


Figure G-1 KC-135 Airplane Simulation

G.1 MATHEMATICAL MODELS

G.1.1 THREE DEGREES OF FREEDOM RIGID BODY VEHICLE DYNAMICS MODEL

The aircraft dynamics are described by the complete nonlinear longitudinal three degrees of freedom [forward (x), vertical (z) and pitch ( )] rigid body vehicle equations of motion written about a fixed vehicle body axis. No mathematical simplifications were made and all nonlinear and coupling terms were included. The three degrees of freedom (DOF) vehicle dynamic model was programmed on the digital computer. The FORTRAN listing of the model is given in Table G-1.

G.1.2 LONGITUDINAL AERODYNAMICS MODEL

The aerodynamic model is used to generate the stability axes longitudinal forces (lift and drag) and moment (pitch) which act on the aircraft. The model and data used in the simulation were taken directly from the KC-135 aerodynamic flight test and simulation document (Reference 3). Lift, drag and moment coefficients were programmed as functions of angle of attack, mach number, stabilizer angle and elevator angle. Spoiler, landing gear and ground effects for approach and landing were also included in the model. The aerodynamic equations were simplified by assuming the rate change of angle of attack and pitch rate are small, there are no flight control failures, and the fuel distribution load factor is zero.

The pilot inputs to the aerodynamic control surfaces used during landing were simulated and included in the aerodynamic model. The active aerodynamic control surfaces simulated in this study were the spoilers (speed brakes), elevator and stabilator.

The aerodynamic model was installed in the digital computer portion of the simulation. The FORTRAN listing of the aerodynamic model is given in Table G-2.

TABLE G-1 THREE DEGREES OF FREEDOM VEHICLE DYNAMICS (SUBROUTINE SG)

```
      SUBROUTINE      SG
C-------------------------------------------------------------------
C  MODEL PACKAGE:      KC-135  FIREPROOF BRAKE HYDRAULIC SYSTEM
C  AUTHOR:             STEVEN M. WARREN
C  DATE:               1/16/81
C
C  THIS SUBROUTINE CONTAINS THE 3 DOF RIGID BODY EQUATIONS OF MOTIONS
C    MODEL WHICH COMPUTES ACCELERATIONS, VELOCITIES AND DISPLACEMENTS
C    OF THE AIRPLANE IN BODY AXIS THEN TRANSFORMS THEM TO THE INERTIAL
C    AXIS
C-------------------------------------------------------------------
C
      INCLUDE "SWGLOBV"
C
C  LINEAR ACCELERATIONS
C
      DELTSG=(FCNT-RNCTSG)*DT/2.0
      UDOT=(XGEAR+FAX-WT*STHTA+FEN)*IMASS-W*Q
      WDOT=(ZGEAR+FAZ+WT*CTHTA)*IMASS+U*Q
C
C  ANGULAR ACCELERATIONS
C
      QDOT=(MA+ME+MGEAR)/IYY
      QDOTD=QDOT*RTOD
C
C  LINEAR VELOCITIES
C
      UTOT=UTOT+(UDOT*3.0-UDOTP)*DELTSG
      WTOT=WTOT+(WDOT*3.0-WDOTP)*DELTSG
      UDOTP=UDOT
      WDOTP=WDOT
      U=UO+UTOT
      W=WO+WTOT
C
C  ANGULAR VELOCITIES
C
      QTOT=QTOT+(QDOT*3.0-QDOTP)*DELTSG
      QDOTP=QDOT
      Q=QO+QTOT
      QD=Q*RTOD
C
C  EULER ANGLES
C
      THTATO=THTATO+(Q*3.0-QP)*DELTSG
      QP=Q
      THTA=THTAO+THTATO
      THTAD=THTA*RTOD
      STHTA=SIN(THTA)
      CTHTA=COS(THTA)
C
```

TABLE G-1 THREE DEGREES OF FREEDOM VEHICLE DYNAMICS (SUBROUTINE SG) CONTINUED

```
C  EARTH TO BODY AXIS ROTATION
C
      B11=CTHTA
      B13=-STHTA
      B31=STHTA
      B33=CTHTA
C
C  WIND VELOCITY COMPONENTS IN BODY AXIS
C
      UWIND=B11*XWIND
      WWIND=B31*XWIND
C
C  INERTIAL AXIS VELOCITIES (EARTH AXIS)
C
      NDOT=B11*U+B31*W
      DDOT=B13*U+B33*W
C
C  CG LOCATION INERTIAL AXIS
C
      XCGTOT=XCGTOT+(NDOT*3.0-NDOTP)*DELTSG
      NDOTP=NDOT
      XCG=XCG0+XCGTOT
      IF(FTDMG.NE.1.) VAR2=XCG
      VAR3=XCG-VAR2
      IF(BRKON.NE.1.0) VAR4=XCG
      VAR5=XCG-VAR4
C
      DTOT=DTOT+(DDOT*3.0-DDOTP)*DELTSG
      DDOTP=DDOT
      ALT=ALTO-DTOT
      IF(U.LE.VAR1) RUNC=1.
      RNCTSG=FCNT
C
C
      RETURN
      END
```

TABLE G-2 AERODYNAMIC MODEL (SUBROUTINE AV)

```
      SUBROUTINE      AV
C-------------------------------------------------------------------
C  MODEL PACKAGE:       KC-135  FIREPROOF BRAKE HYDRAULIC SYSTEM
C  AUTHOR:              STEVEN M. WARREN
C  DATE:                1/16/81
C
C  THIS SUBROUTINE CALCULATES THE AERODYNAMIC COEFFICIENTS, AIRSPEEDS,
C    FORCES AND MOMENTS USED IN THE 3 DOF AIRPLANE SIMULATION.  THE
C    COEFFICIENTS ARE CALCULATED IN STABILITY AXIS AND RESOLVED INTO
C    BODY AXIS.
C-------------------------------------------------------------------------
      INCLUDE "SWGLOBV"
C
C  AIR SPEEDS AND ANGLE OF ATTACK
C

      UA=U-UWIND
      WA=W-WWIND
      VPSQ=UA*UA+WA*WA
      QBAR=RHO2*VPSQ
      ALFAAP=ATAN(WA/UA)
      ALFAWG=ALFAAP*RTOD+ALWING
      ALFA=ALFAWG*DTOR
      VP=SQRT(VPSQ)
      VE=VP*SQRT(147.7*RHO)
      VE2=VE*VE
      VE3=VE2*VE
      MACH=VP/A
      MACH2=MACH*MACH
      MACH3=MACH2*MACH
      SALFA=SIN(ALFAAP)
      CALFA=COS(ALFAAP)
      AL1=ALFAWG
      AL2=AL1*AL1
      AL3=AL2*AL1
      AL4=AL3*AL1
      AL5=AL4*AL1
C
C  STABILIZER DELECTION
C
      STAB=STABO
C
C   SPOILER (SPEED BRAKES) DEFLECTION
C
      SP=-SPON*(SPO-SPF)+SPO
      SP2=SP*SP
      SP3=SP2*SP
      SP4=SP3*SP
      SP5=SP4*SP
```

TABLE G-2 AERODYNAMIC MODEL (SUBROUTINE AV)
CONTINUED

```
C
C  ELEVATOR DEFLECTION
C
      ELEV=-THTAC*(ELEVO-ELEVF)+ELEVO
      ELEV2=ELEV*ELEV
      ELEV3=ELEV2*ELEV
C
C  BASIC AERODYNAMIC COEFFICIENTS (LONGITUDINAL)
C
C   LIFT
C
      CLB=-0.00009428388*AL3+0.0003664335*AL2+0.08183358*AL1+0.768162
      CLV=0.0000000005144315*VE3-0.000004027201*VE2+0.0000764562*VE+
     X     0.000__34807
      CLA=(0.0000000002906583*VE3-0.000000234813*VE2+0.000002183605*VE-
     X      0.00003138558)*AL1
      KALFA=1.0
      IF(ALFAWG.LT.0.0) KALFA=1.0+0.025*ALFAWG
      IF(ALFAWG.GT.5.0) KALFA=1.0755-0.0131*ALFAWG
      KELEV=0.0000009824306*ELEV3-0.0002004688*ELEV2-0.000710069*ELEV+
     X      1.00303
      CLS=(STAB+6.0)*KALFA*(0.005725784*MACH3-0.00849203*MACH2-
     X       0.0001242563*MACH+0.009986454)
      CLE=KALFA*KELEV*ELEV*(0.003483229*MACH3-0.007722376*MACH2-
     X       0.0001811669*MACH+0.00431884)
      CLSP=(0.00009079838*AL3-0.0001006855*AL2-0.004219678*AL1-0.336252)*
     X     2.0*(SP/60.0)**0.3333
      CLGRD=0.049+0.004*AL1
      CLA1=CLB+CLV+CLA+CLS
      CLA2=CLA1*CLA1
      CLA3=CLA2*CLA1
      CL=CLA1+CLE+CLSP+CLGRD
C
C   DRAG
C
      CDB=0.08462839*CLA3-0.139644*CLA2+0.129189*CLA1+0.06652674
      CDSP=((-0.00000000002896914*SP5+0.00000001817408*SP4-
     X     0.000002529863*SP3+0.0001401342*SP2-0.002761208*SP-
     X     0.0005764344)+(-0.00000000002036258*SP5+0.000000004160344*SP4-
     X     0.000000333125*SP3+0.00001290896*SP2-0.0002424279*SP-
     X            0.00004075552)*(ALFAWG-4.0))*2.0
      CDLG=0.023
      CDGRD=0.00001034123*AL5-0.0000482657*AL4-0.0002135557*AL3+
     X      0.001247322*AL2-0.001009659*AL1-0.04159739
      CD=CDB+CDSP+CDLG+CDGRD
C
C
C   PITCH
C
      CMB=-0.000008208773*AL3+0.000677266*AL2-0.02133708*AL1-0.0271406
```

TABLE G-2 AERODYNAMIC MODEL (SUBROUTINE AV)

CONTINUED

```
      CMV=-0.0000000012033*VE3+0.0000008450631*VE2-0.00003600811*VE-
     X    0.000005718992
      CMA=(-0.0000000001044415*VE3+0.000000103761*VE2-0.0000000589349B*VE-
     X      0.00001405401)*AL1
      CMCG=CL*(CG-0.25)
      CME=KALFA*KELEV*ELEV*(-0.010546*MACH3+0.02267096*MACH2-
     X       0.00009310627*MACH-0.01303612)
      CMS=KALFA*(STAB+6.0)*(-0.02486061*MACH3+0.03268786*MACH2-
     X    0.00141173*MACH-0.02988064)
      CMSP=(0.0000007091581*AL5-0.00001281677*AL4-0.000005324615*AL3+
     X      0.0002525657*AL2+0.0009433642*AL1+0.08659211)*2.0*
     X      (0.00000002421542*SP5-0.000004274382*SP4+0.00028493*SP3-
     X      0.00893891*SP2+0.13686*SP+0.02336269)
      CMLG=-0.003
      CMGRD=0.0162078*CLA3-0.005471269*CLA2-0.09849432*CLA1+0.006799686
      CM=CMB+CMV+CMA+CMCG+CME+CMS+CMSP+CMLG+CMGRD
C
C
C  BODY AXIS COEFFICIENTS
C
      CX=CL*SALFA-CD*CALFA
      CZ=-CL*CALFA-CD*SALFA
C
C  AERODYNAMIC FORCES
C
      QBARAW=QBAR*AW
      FAX=CX*QBARAW
      FAZ=CZ*QBARAW
C
C  AERODYNAMIC MOMENTS
C
      MA=CM*QBARAW*CHORD
C
C
      RCNTAV=FCNT
C
C
      RETURN
      END
```

G.1.3 LANDING GEAR MODEL

The landing gear system model used in the KC-135 hybrid simulation contains two degrees of freedom; compression and fore-aft motion. The compression degree of freedom is required to determine the vertical load on the landing gear and tire The vertical force is a function of the combined shock strut and tire stroke and stroke rate. The fore-aft degree of freedom is required to assess landing gear stability (coupling between the antiskid system, brake hydraulic system and landing gear). The compression degree of freedom was programmed on the digital computer while the fore-aft freedom was programmed on the analog computer. A FORTRAN listing of the compression DOF is given in Table G-3. A block diagram of analog computer of the fore-aft DOF is given in Figure G-2.

G.1.4 GROUND FORCE MODEL

The ground force model is used to generate the main gear tire ground force. The ground force is a nonlinear function of tire slip, runway friction and vertical tire load. This nonlinear relationship is given in Figure G-3. The primary portion of the ground force model was programmed on the analog computer, however precalculation of several key variables was performed on the digital computer. A block diagram of the analog computer portion of the simulation is given in Figure G-4 while a FORTRAN listing of the digital computer portion is given in Table G-4.

G.1.5 BRAKE TORQUE MODEL

The brake torque model defines the nonlinear relationship between brake pressure and brake torque. The model converts the brake pressure (from brake hydraulic system mockup) to brake torque for use in the calculation of wheel speed. The model includes the static pressure-torque gain characteristic, the dynamic torque response relationship, torque peaking and torque fade characteristics of the brake. The brake torque model was programmed entirely on the analog computer. A block diagram of the model is given in Figure G-5.

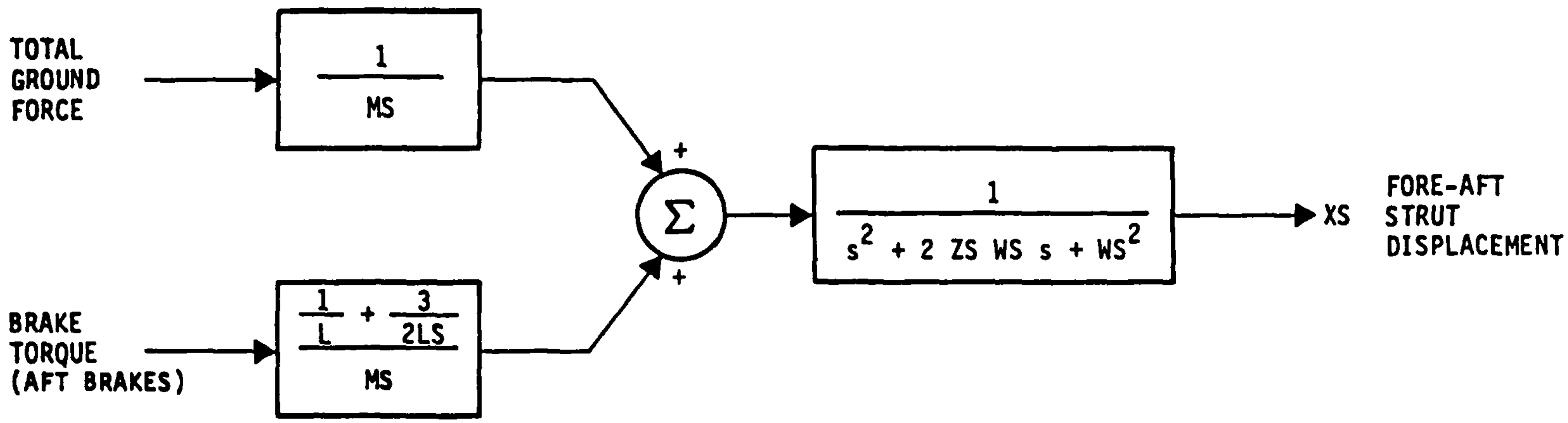
TABLE G-3 LANDING GEAR MODEL, COMPRESSION (SUBROUTINE OLEOS)

```
      SUBROUTINE      OLEOS
C-----------------------------------------------------------------
C  MODEL PACKAGE:        KC-135  FIREPROOF BRAKE HYDRAULIC SYSTEM
C  AUTHOR:               STEVEN M. WARREN
C  DATE:                 1/16/81
C
C  THIS SUBROUTINE DETERMINES THE VERTICAL GROUND FORCE AT THE TIRE
C    RUNWAY INTERFACE.  THE UNSPRUNG MASS HAS BEEN ELIMINATED.  OLEO
C    STROKE AND RATE ARE DETERMINED AT THE GEAR ATTACHMENT POINT.
C-----------------------------------------------------------------
      INCLUDE "SWGLOBU"
C
C  PRECALCULATIONS
C
C    NOSE
C
      ZAN=(ALT-(B13*XNG+B33*ZNA))*12.0
      HNG=B33*HN
      XPN=(HNG-ZAN)/B33
      IF(ZAN.GE.HNG) XPN=0.0
      IF(FTDNG.EQ.1.) GO TO 1
      IF(ZAN.GE.HNG)  GO TO 1
      FTDNG=1.
      CALL CDIO
 1    XDPN=(DDOT+(-XNG*Q))*12.0
      IF(XPN.LE.0.0) XDPN=0.0
C
C    MAIN
C
      ZAM=(ALT-(B13*XMG+B33*ZMA))*12.0
      HMG=B33*HM
      XPM=(HMG-ZAM)/B33
      IF(ZAM.GE.HMG) XPM=0.0
      IF(FTDMG.EQ.1.) GO TO 2
      IF(ZAM.GE.HMG)  GO TO 2
      FTDMG=1.
      CALL CDIO
 2    XDPM=(DDOT+(-XMG*Q))*12.0
      IF(XPM.LE.0.0) XDPM=0.0
C
C  OLEO ORIFICE DISCHARGE COEFFICIENT
C
      AOMPM=TLXY(XPM,XOMG,AOMG,NXOMG)
      AOMPN=TLXY(XPN,XONG,AONG,NXONG)
C
      C1N=0.5*RHOO*APNG*APNG*APNG/
     X    (CDNG*CDNG*AOMPN*AOMPN)
      C1M=0.5*RHOO*APMG*APMG*APMG/
     X    (CDMG*CDMG*AOMPM*AOMPM)
```

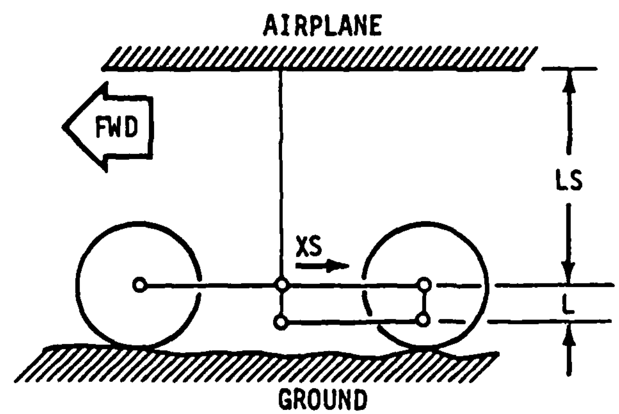
TABLE G-3 LANDING GEAR MODEL, COMPRESSION (SUBROUTINE OLEOS)

CONTINUED

```
C
C  OLEO FORCE CALCULATIONS (VERTICAL GROUND FORCE)
C
C
C    NOSE
C
      AXDN=ABS(XDPN)
      FAN=TLXY(XPN,XPNG,FANG,NXPNG)
      FMPN=C1N*XDPN*AXDN
      FFRN=KLDN*XDPN
      FFN=FFRN
      IF(FFRN.GE.FFNUL) FFN=FFNUL
      IF(FFRN.LE.FFNLL) FFN=FFNLL
      FSNG=-(FAN+FMPN+FFN)
C
C    MAIN
C
      AXDM=ABS(XDPM)
      FAM=TLXY(XPM,XPMG,FAMG,NXPMG)
      FMPM=C1M*XDPM*AXDM
      FFRM=KLDM*XDPM
      FFM=FFRM
      IF(FFRM.GE.FFMUL) FFM=FFMUL
      IF(FFRM.LE.FFMLL) FFM=FFMLL
      FSMG=-(FAM+FMPM+FFM)
C
C
      RETURN
      END
```



L = TORQUE MOMENT ARM
LS = STRUT LENGTH
MS = STRUT MASS
s = LAPLACE OPERATOR
WS = FORE-AFT STRUT NATURAL FREQUENCY
XS = FORE-AFT STRUT DISPLACEMENT
ZS = FORE-AFT STRUT DAMPING RATIO



KC-135 MAIN LANDING GEAR TRUCK

Figure G-2 KC-135 Landing Gear Model, Fore-Aft Degree of Freedom

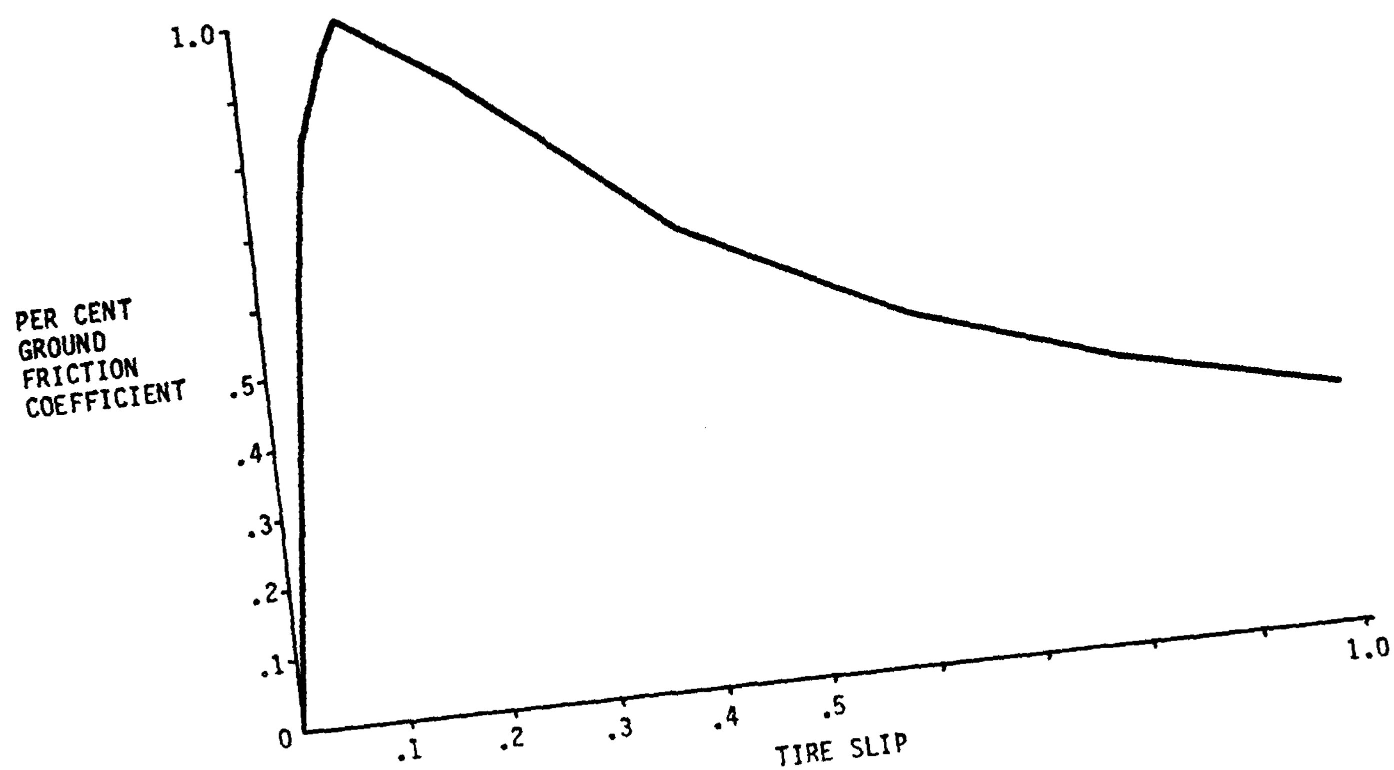
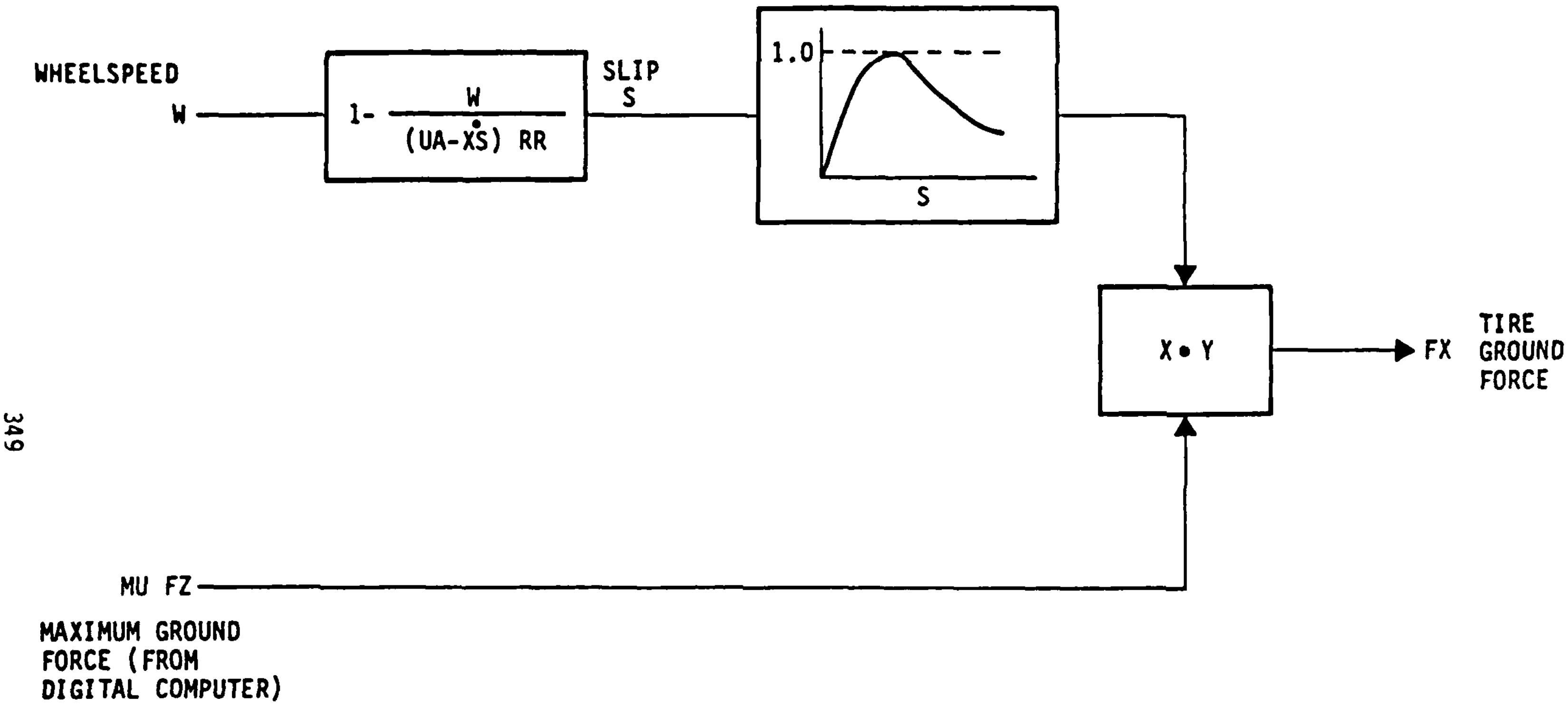


Figure G-3 MU-Slip Ground Force Relationship



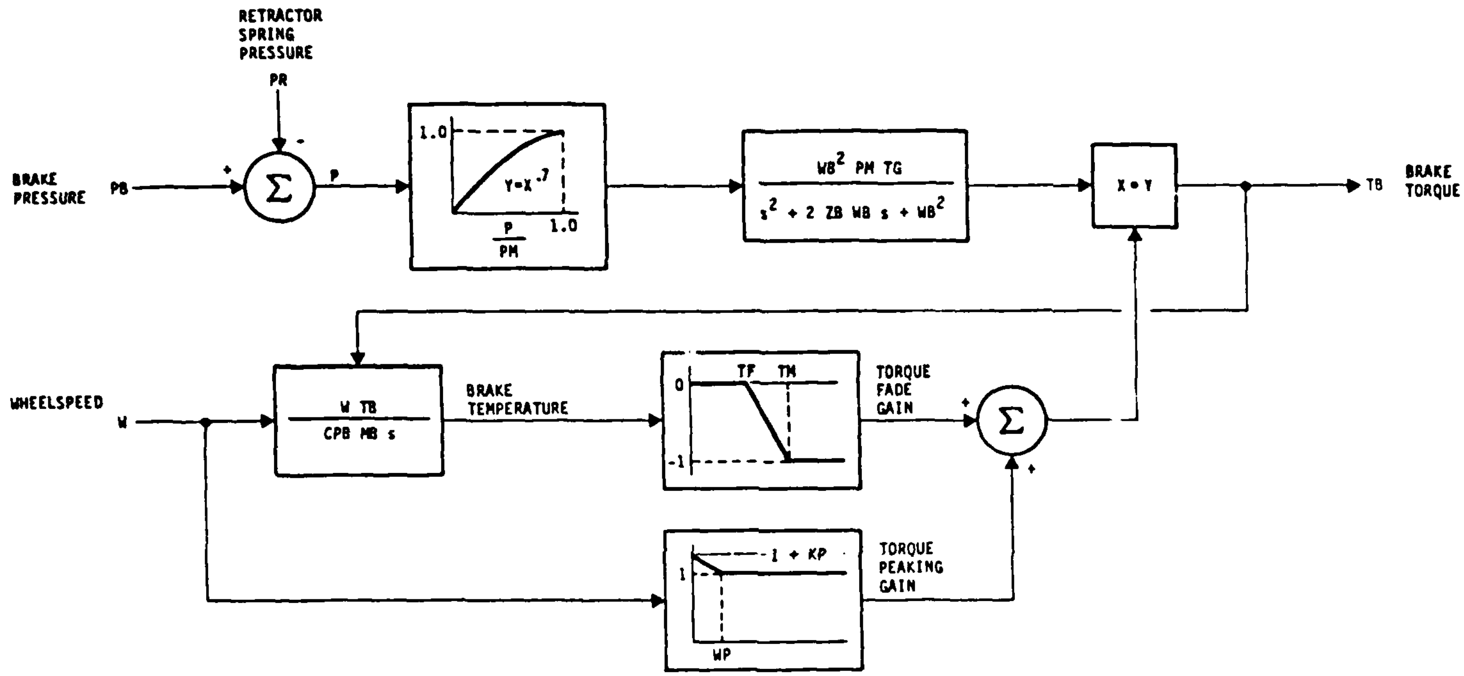
FX	TIRE GROUND FORCE	S	TIRE SLIP
FZ	VERTICAL TIRE LOAD	UA	FORWARD VELOCITY OF AIRPLANE
MU	RUNWAY FRICTION	W	TIRE WHEELSPEED
RR	TIRE ROLLING RADIUS	$\dot{X}S$	FORE-AFT STRUT VELOCITY

Figure G-4 Ground Force Model

TABLE G-4 GROUND FORCE MODEL (SUBROUTINE GFORCE)

```
      SUBROUTINE      GFORCE
C-----------------------------------------------------------------
C  MODEL PACKAGE:      KC-135  FIREPROOF BRAKE HYDRAULIC SYSTEM
C  AUTHOR:             STEVEN M. WARREN
C  DATE:               1/16/81
C
C  THIS SUBROUTINE CONTAINS OR CALCULATES
C      1.  PARTIAL MAIN GEAR GROUND FORCE MODEL
C      2.  AXLE VELOCITY (EXCLUDING GEAR WALK) - MAIN GEAR
C      3.  GROUND FORCE MOMENT ARMS
C-----------------------------------------------------------------

      INCLUDE "SWGLOBV"
C
      DELTGF=(FCNT-RCNTGF)*DT/2.0
C
C  GROUND FORCE MODEL (PRECALCULATIONS)
C
C    MAIN
C
      ZMR=ZMGS-XPM/12.0
      ZMRA=ZMR-TRMG/12.0
      UAMR=U+ZMRA*Q
C
C  GROUND FORCE MODEL
C
C    MAIN
C
      ABSFZM=ABS(FSMG/NBM)
      MUMFZ=MU(IMUV)*(1.0-K3*ABS(UAMR))*ABSFZM
      FXL=0.0
      FXR=0.0
      IF(BRKON.EQ.1.0) FXL=MUMFZ
      IF(BRKON.EQ.1.0) FXR=MUMFZ
C
      RCNTGF=FCNT
C
C
      RETURN
      END
```



CPB	SPECIAL HEAT OF BRAKE MATERIAL	PR	RETRACTOR SPRING PRESSURE	W	WHEELSPEED
KP	TORQUE PEAKING GAIN	s	LAPLACE OPERATOR	WB	NATURAL FREQUENCY OF TORQUE RESPONSE
MB	BRAKE HEAT SINK MASS	TB	BRAKE TORQUE	WP	WHEELSPEED AT INITIATION OF TORQUE PEAKING
P	EFFECTIVE PRESSURE	TF	TEMPERATURE AT INITIATION OF BRAKE FADE	ZB	DAMPING RATIO OF TORQUE RESPONSE
PB	BRAKE PRESSURE	TG	PRESSURE-TORQUE GAIN		
PM	MAXIMUM BRAKE PRESSURE	TM	MAXIMUM BRAKE TEMPERATURE		

Figure G-5 Brake Torque Model

G.1.6 WHEEL DYNAMICS MODEL

The wheel dynamics model is used to calculate the rotational velocity of the wheel. Inputs to the model are ground force and brake torque. The output wheel speed signal is used by the antiskid system for control of brake pressure. The wheel dynamics were implemented on the analog computer portion of the hybrid simulation due to the high resonant frequencies associated with wheel spin up and skidding (braking activity). A block diagram of the wheel dynamics model is given in Figure G-6.

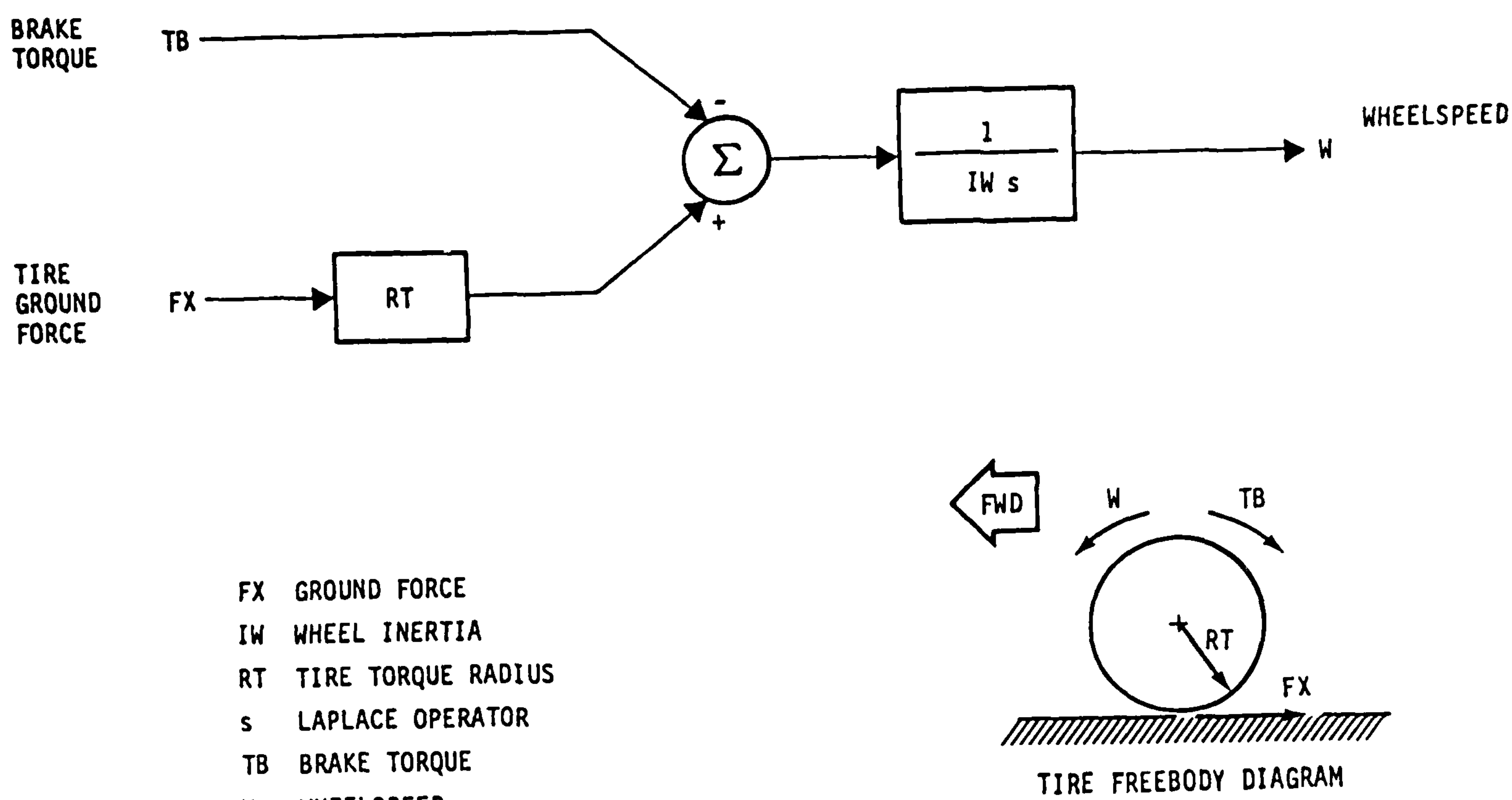
G.1.7 FORCE RESOLUTION CALCULATIONS

A series of calculations are included in the simulation to determine the total body axes (longitudinal) forces and moment due to externally applied ground forces. Contained in these calculations are transformation matrices which rotate the vertical and ground forces at each landing gear from the inertial axes into the aircraft body axes. The resulting total applied body axes forces and moment are inputs to the three degrees of freedom rigid body vehicle equations of motion. The FORTRAN listing of the digital computer subroutine, RESOLV, which contains these calculations is given in Table G-5.

Also, contained in the RESOLV subroutine is the engine thrust model. Engine thrust has been programmed as a function of mach number. Provisions for adjusting the thrust for trim flight at the start of a run and pilot inputs during flight and landing are included. It was assumed that the KC-135 engine thrust acts parallel to the aircraft body axis.

G.2 HARDWARE

A KC-135 brake hydraulic system mockup, a two-fluid brake hydraulic system and a mockup KC-135 Mark II antiskid control box were used in the simulation. The mockups were built with actual KC-135 aircraft hardware supplied by the Air Force. The hydraulic system mockups are described in detail in Appendix E.



FX GROUND FORCE
IW WHEEL INERTIA
RT TIRE TORQUE RADIUS
s LAPLACE OPERATOR
TB BRAKE TORQUE
W WHEELSPEED

Figure G-6 Wheel Dynamic Model

TABLE G-5 FORCE RESOLUTION CALCULATIONS (SUBROUTINE RESOLV)

```
      SUBROUTINE      RESOLV
C-----------------------------------------------------------------
C  MODEL PACKAGE:      KC-135 FIREPROOF BRAKE HYDRAULIC SYSTEM
C  AUTHOR:             STEVEN M. WARREN
C  DATE:               1/16/81
C
C  THIS SUBROUTINE CALCULATES THE TOTAL FORCE AND MOMENT AT THE
C    AIRPLANE CENTER OF GRAVITY (IN BODY AXIS) DUE TO THE EXTERNAL
C    FORCES APPLIED AT EACH GEAR.  IN ADDITION, THE FORCE AND MOMENT
C    (IN BODY AXIS) DUE TO THE ENGINE IS DETERMINED.
C------------------------------------------------------------------
C
      INCLUDE "SWGLOBV"
C
C  GEAR FORCES
C
C
C    MAIN RIGHT
C
      FRXR=0.0
      IF(FXR.GE.DB1) FRXR=-(FXR-DB1)*NBM
      IF(FXR.LE.-DB1) FRXR=-(FXR+DB1)*NBM
C
C    MAIN LEFT
C
      FRXL=0.0
      IF(FXL.GE.DB1) FRXL=-(FXL-DB1)*NBM
      IF(FXL.LE.-DB1) FRXL=-(FXL+DB1)*NBM
C
C  TOTAL APPLIED FORCE
C
      XGEAR=FRXR+FRXL
      ZGEAR=FSNG+2.0*FSMG
C
C  GEAR MOMENTS
C
C    NOSE
C
      TYN=-XNG*FSNG
C
C    MAIN RIGHT
C
      TYR=ZMR*FRXR-XMG*FSMG
C
C    MAIN LEFT
C
      TYL=ZMR*FRXL-XMG*FSMG
C
C  TOTAL APPLIED MOMENT
C
```

TABLE G-5 FORCE RESOLUTION CALCULATIONS (SUBROUTINE RESOLV)

CONTINUED

```
      MGEAR=TYN+TYR+TYL
C
C  ENGINE THRUST MODEL - THRUST AS A FUNCTION OF MACH NUMBER
C
C    THRUST - ONLY ALONG X BODY AXIS
C
      IF(ENCUT.GT.0.0) PCTH=PCTH*(1.0-(FCNT-EFCNT)*DT/TAUE)
      FEN=(FENO*(1+KTRM)+(FENMAC*MACH)
     X          -(FENO*(1+KTRM)-FENOI)*(1-PCTH))*NENG
C
C    MOMENT
C
      ME=ZENG*FEN
C
      EFCNT=FCNT
C
      RETURN
      END
```

G.3 AIRPLANE SIMULATION SETUP

The mathematical models, hydraulic mockup, and antiskid control box were integrated and implemented in the HYBCOL to form the real time KC-135 braking simulation. To make maximum use of the HYBCOL's capability the mathematical models were implemented as follows:

Models Installed on the Digital Minicomputer:

- o Three DOF rigid body vehicle dynamcis
- o Longitudinal aerodynamics
- o Landing gear (compression)
- o Ground force (precalculations)
- o Force resolution calculations

Models Installed on the Analog Computers:

- o Ground force
- o Brake torque
- o Wheel dynamics
- o Landing gear (fore and aft)

Figure G-1 is a schematic detailing the manner in which the mathematical models, antiskid control box and hydraulic system mockup were integrated to form the aircraft simulation. Only one landing gear and brake system is shown in the figure for simplicity.

G.4 KC-135 DATA

Engineering data concerning the KC-135 configuration, weight, inertias, and aerodynamics was taken directly from Air Force and Boeing Aircraft Company reports (References 1, 3, 5, 6 and 7).

The data used in the simulation is compiled in Appendix J.

G.5 BASELINE AIRCRAFT

A baseline KC-135 aircraft configuration was defined for the brake system performance evaluation. A 150,000 pound gross weight KC-135 aircraft in a landing configuration, with gear down was selected for the baseline aircraft. This baseline description represents a KC-135 in a typical aircraft landing configuration. Complete weight, inertia, and aerodynamic trim data was obtained from Reference 3.

Aerodynamic coefficients consistent with the KC-135 landing procedures were programmed in the aerodynamic model. The KC-135 Flight Manual (Reference 6) specifies that 50 degrees flaps and a trim stabilizer be used during the approach and landing, and at touchdown the stick (elevator control) be moved to the full forward position. Aerodynamic data consistent with this configuration was taken from Reference 3.

The basic baseline aircraft configuration data is listed in Table C-6. Additional airplane data is listed in Appendix J.

G.6 TRIM AIRCRAFT

Each simulation run was initiated with the aircraft trimmed in the air. The initial aircraft operating conditions (sink speed, forward velocity, and altitude) are specified. Trim values of engine thrust, angle of attack and stabilizer angle are determined at this operating point such that the pitch acceleration, pitch rate and linear accelerations are zero.

G.7 SIMULATION OF PILOT INPUTS DURING LANDING

During landing the pilot makes inputs to the engines, brakes, spoilers and elevator to control the aircraft's attitude and deceleration. These pilot inputs were programmed as simple time dependent functions and in the following sequence.

TABLE G-6 KC-135 DATA (FILE NAME KCDATA)

```
;C-----------------------------------------------------------------
;C  FILE NAME:            KCDATA
;C  MODEL PACKAGE:        KC-135 FIREPROOF BRAKE HYDRAULIC SYSTEM
;C  AUTHOR:               STEVEN M. WARREN
;C  DATE:                 1/16/81
;C
;C  THIS FILE CONTAINS THE KC-135 INPUT DATA USED IN THE DIGITAL
;C    COMPUTER SIMULATION
;C-------------------------------------------------------------------
;
;  BASIC RTE PARAMETERS
;
MU(1)=0.600     =.500   =.400   =.300   =.200   =.100
MU(7)=.075 =.050 =1. =1.
IMUV=1


;
;  INPUT DATA PARAMETERS
;
;  SHOCK STRUT/OLEO DATA   (OLEOS)
;
        APMG  =68.67,   CDMG  =0.88,    KLDM  =100.0,
        APNG   =15.41,  CDNG  =0.88,    KLDN  =100.0,
        FFMLL=-1000.,   FFMUL=1000.,    FFNLL=-1000.0,  FFNUL=1000.,
        LMG   =89.190,  LNG   =55.70,   RHOO =0.0000795
;
;  AIRCRAFT CONFIGURATION DATA   (SG)
;
        WT=150000.,     IYY=2410000., VPO   =204.0,  DDOTO  =-3.0,
        WLENG=193.6,    WLCG=193.6,   WLMGA=177.6,   WLNGA=147.16,
        FSLMAC=786.17,  FSMGA=887.0,  FSNGA=339.0,   GVTY  =32.174,
;
;  ENGINE DATA   (RESOLV)
;
        FENMAC =-1486.2,   FENO    =600.0,   FENOI  =600.0,   NENG   =4.0
        TAUE   =2.0
;
;  TIRE DATA   (GFORCE)
;
        KMT  =11781.0,  KNT    =7388.0,   K3     =0.0,
        NBM    =4.0,    RTM  =24.15,      TRMG   =22.29,
        NBN    =2.0,    RTN  =19.0,       DB1   =15.0,
;
;  AERODYNAMIC DATA   (AV)
;
;    THE LIFT, DRAG AND MOMENT CEFFICIENTS INCLUDE GROUND EFFECTS
;
        XWIND =0.0,   MAC   =241.8,    CG    =0.296, AW     =2433.0,
        ALWING=2.0,   RHO   =0.002377,
;
```

TABLE G-6 KC-135 DATA (FILE NAME KCDATA)
CONTINUED

```
; FLAGS   (LFLAG=0  AIRCRAFT IN AIR, ENCUT=1 ENGINE THROTTLES CUT AT T/D)
;
     LFLAG  =0.0,    ENCUT  =0.0
;
; TRIM/LANDER
;
     DT     =0.015,  DTSP  =2.0,     DTELEV=2.0,
     EPSWD  =.001,   DH    =-.2,     DALFA  =.1,     ALFAT  =0.0
     EPSUD  =0.01,   DKTRM =-0.1,    SPO  =0.0,      SPF=60.0,
     EPSQD  =.001,   DSTAB =0.1,     ELEVO   =0.0,   DALFAG =.1
     ELEVF  =+15.0,  STABO =0.0,     DTBRK  =2.0,    DTENG=2.0

;
; INITIAL CONDITONS
;
     ALTO   =20.0,   XCGO   =0.0,
     QO    =0.0,     QDOTP =0.0,   UDOT  =0.0,   WDOT  =0.0,
;
; INPUT TABLES
;
;   METERING PIN ORIFICE AREA
;
;     MAIN GEAR
;
     NXOMG=8
     XOMG(1)=0.0     =1.30    =7.93    =9.05    =10.7    =11.55   =21.80
     XOMG(8)=22.0
     AOMG(1)=3.2326  =3.2326  =1.8007  =1.8007  =2.0091  =2.0091  =.0796
     AOMG(8)=.0796
;
;     NOSE GEAR
;
     NXONG=10
     XONG(1)=0.0     =.156    =5.566   =9.036   =12.536   =13.036   =15.096
     XONG(8)=15.346  =15.846  =16.00
     AONG(1)=.4304   =.4304   =.2879   =.2879   =.2169   =.2169   =.0496
     AONG(8)=.0496   =.01     =.01
;
;  OLEOS     COMBINED SHOCK STRUT AND TIRE LOAD DEFLECTION CURVES
;
;     NOSE GEAR
;
     NXPNG  =10
     XPNG(1)=0.     =5.15    =10.875    =12.5   =13.925   =15.2   =16.4
     XPNG(8)=17.5   =18.925  =19.9
     FANG(1)=0.     =3926.   =6870.   =8834.  =11778.  =15704.  =21593.
     FANG(8)=29445   =43186.  =54964.
;
;     MAIN GEAR
;
```

TABLE G-6 KC-135 DATA (FILE NAME KCDATA)

CONTINUED

```
        NXPMG  =10
        XPMG(1)=0.      =5.8    =13.3   =16.06  =19.03  =20.85  =22.45
        XPMG(8)=23.90   =25.05    =26.35
        FAMG(1)=0.      =19635. =31416. =39270. =54978. =70686. =90321.
        FAMG(8)=117810.  =141372.  =180642.
;
;
;       GENERAL USE PARMS
;
VAR1=24.0        ; COMPUTER CUT OUT SPEED

 : 3- 5-80   14: 8:23
 HZERO=10

 : 5- 2-80   13:31:42
 RVON=1

 :12- 1-80   14:27:35
 DTELEV=0.0

 :12- 1-80   14:47:53
 KROT=.5

 :12- 4-80   13:12:14
 KSP=.5
```

Spoiler deployment	0.25 second after touchdown
Stick forward (elevator)	1.00 second after touchdown
Cut engine throttles	1.50 seconds after touchdown
Brake application	2.00 seconds after touchdown

The FORTRAN listing containing the pilot input function is given in Table G-7.

G.8 COMPUTER SIMULATION TO AIRCRAFT CORRELATION

The KC-135 hybrid computer simulation was correlated with actual aircraft flight test data to verify the aircraft models, input data and overall simulation. The effort included correlation of aerodynamic trim data and braking performance.

G.8.1 AERODYNAMIC TRIM DATA CORRELATION

The aerodynamic trim correlation effort was performed to verify the rigid body airplane model and aerodynamic input data. Data from Reference 3 was used for the correlation effort. The airplane was trimmed in the air (at several flight conditions) using the hybrid computer simulation. The trim angle of attack, engine thrust and stabilizer angle were compared with the flight test data. Excellent correspondence was achieved between the data and the simulation.

G.8.2 BRAKING PERFORMANCE CORRELATION

The braking correlation was conducted with data obtained from Combat Traction II, Phase II Technical Report Volumes I (Reference 8) and II (Reference 1). No attempt was made to duplicate airplane stopping distance exactly. Instead, emphasis was placed on producing the same brake system control characteristics. Stopping distances were matched by adjusting the tire-to-ground friction coefficient. Other parameters were then checked and, if necessary, adjusted to improve correlation with flight test records. The skid pressure level, depth of skid, rates in and out of skids, and the number of skids per second were evaluated to ensure that the tire and ground force simulation was properly adjusted.

TABLE G-7 PILOTS INPUTS (SUBROUTINE LANDER)

```
      SUBROUTINE      LANDER
C ---------------------------------------------------------------
C MODEL PACKAGE:      KC-135  FIREPROOF BRAKE HYDRAULIC SYSTEM
C AUTHOR:                     R.L. AMBERG
C DATE:                       3/79
C REVISION:                   1.03    1/16/81   S. M. WARREN
C
C     THE LANDING SCHEDULE MODEL PROVIDES THE SCHEDULING OF EVENTS
C     DURING A SIMULATED LANDING/STOP.
C ---------------------------------------------------------------

      INCLUDE 'SWGLOBV'

      IF(FTDMG.EQ.1.)GO TO 2       ; MAIN GEAR TOUCHDOWN
      LTTD = 0
      GO TO 3

2     LTTD = LTTD+(FCNT-LFCNT)*DT
C
C DEROTATION COMMAND / ELEVATOR CONTROL
C
3     THTAC=0.0
      THTAC = KROT*(LTTD-DTELEV)
      IF (THTAC.LE.0.0) THTAC=0.0
      IF (THTAC.GE.1.0) THTAC=1.0
C
C SPOILER CONTROL
C
      SPON=0.0
      SPON=KSP*(LTTD-DTSP)
      IF(SPON.LE.0.0) SPON=0.0
      IF(SPON.GE.1.0) SPON=1.0
C
C ENGINE CUT
C
      ENCUT=0.0
      IF(LTTD.GT.DTENG) ENCUT=1.0

C
4     IF(RVON.EQ.1.0) GO TO 5
      IF(RVON.EQ.2.0) GO TO 40
C
      GO TO 5
C
40    IMUV=2
      MUTIME=MUTIME+DT*(FCNT-LFCNT)
      IF(MUTIME.GE.1.0) IMUV=8
      IF(MUTIME.GE.1.5) MUTIME=0.0
C
```

TABLE G-7 PILOTS INPUTS (SUBROUTINE LANDER)
CONTINUED

```
5       IF(BRKON.NE.0.)         GOTO 6          ; BRAKES
        IF(LTTD.LE.DTBRK)       GOTO 6
        BRKON=1.
        CALL CDIO

6       LFCNT=FCNT                      ;UPDATE LANDER FRAME COUNT.

        RETURN
        END
```

APPENDIX H

FLUID SAMPLES

CTFE fluid samples were taken during the component and system tests. These fluid samples were submitted to the Air Force for analysis. The sampling history is given in Table H-1. Results of the Air Force analysis of the fluid samples are givenin Table H-2.

The fluid samples were submitted to the Air Force in bottles supplied by the Boeing Fuels and Lubricants Laboratory. The bottles were cleaned before use in the following manner:

(1) Rinsed with freon
(2) Washed with soap and distilled water
(3) Rinsed with distilled water
(4) Evaculated dry
(5) Cap seals were replaced with new telfon seals
(6) Bottles were capped and stored in plastic bags.

A sample of CTFE fluid (taken directly from an AFWAL container) was analyzed by the Boeing Fuels and Lubricants Laboratory. The kinematic viscosity of the CTFE fluid at 100 degrees Fahrenheit was found to be 3.3 centistokes. The fluid acid number of the sample was 0.07 milligrams of KOH/gram of sample fluid.

TABLE H-1 FLUID SAMPLES

SAMPLE NUMBER	DESCRIPTION	COMMENT	DATE	TEST TEMP. °F	SOAK TIME AT TEMP. HR & MIN	TEST TIME HR & MIN	TOTAL TEST TIME ON SAMPLE HR & MIN
0	New CTFE fluid	Unused fluid taken from AFWAL container	2-10-81				0
	Testing	Modified brake component testing	2-13-81	Ambient	--	4:12	
			2-16-81	-65	6:30	1:06	
			2-17-81	160	7:10	:55	
1	End of modified brake component testing	Fluid removed from brake bleed port	2-18-81				6:13
	Testing	Modified deboost valve component testing	3-16-81	Ambient	--	5:17	
			3-20-81	-65	6:18	1:55	
			3-23-81	160	6:15	2:05	
2	End of modified deboost valve component testing	Fluid removed from Port B, Green contaminant appears in fluid	3-24-81				9:17
	Testing	Two-fluid system tests System serviced with CTFE 1 gallon CTFE added to pump reservoir					

TABLE H-1 FLUID SAMPLES (CONTINUED)

SAMPLE NUMBER	DESCRIPTION	COMMENT	DATE	TEST TEMP. °F	SOAK TIME AT TEMP. HR & MIN	TEST TIME HR & MIN	TOTAL TEST TIME ON SAMPLE HR & MIN
3a	Start of two-fluid system tests	Fluid removed from brake bleed port during servicing	4-13-81				0
	Testing	Frequency and step response tests	4-15-81	Ambient	--	2:45	
3b	Sample		4-15-81				2:45
	Testing	Complete ambient temp. tests	4-15-81	Ambient	--	1:15	
3c	Conclusion of ambient temp. system tests	System serviced, fluid removed from brake	4-15-81				4:00
	Testing	System tests at 160°F, complete test series performed	4-16-81	160	7:05	3:45	
3d	Conclusion of high temp. system temps	System serviced, fluid removed from brake Green contaminant in fluid	4-16-81				

TABLE H-1 FLUID SAMPLES (CONTINUED)

SAMPLE NUMBER	DESCRIPTION	COMMENT	DATE	TEST TEMP. °F	SOAK TIME AT TEMP. HR & MIN	TEST TIME HR & MIN	TOTAL TEST TIME ON SAMPLE HR & MIN
	Testing	System tests at -40°F, complete test series performed	4-17-81	-40	6:15	4:05	
3e	Conclusion of -40°F system tests	System serviced, fluid removed from brake Green contaminant in fluid	4-20-81				11:50
	Testing	1 gallon CTFE added to fluid reservoir	5-4-81	-65	6:45	3:30	
		System tests at -65°F					
3f	Conclusion of -65°F system tests	Fluid removed from brake	5-6-81				11:50

TABLE H-2 AIR FORCE ANALYSIS OF FLUID SAMPLES

SAMPLE NUMBER	WATER DETERMI-NATIONS (ppm)	VISCOSITY @ 38°(100°F) (cs)	VISCOSITY CHANGE (%)	SATURATION	
				WITHOUT ACID WASH	WITH ACID WASH
0	50	3.30	--	1 hr	--
1	85	3.33	+1	60 min	--
2	46	3.36	+2	60	--
3a	69	3.34	+1	60	--
3b	65	3.32	+1	60	--
3c	28	3.36	+2	60	--
3d	220	3.28	-1	60	--
3e*	390	3.32	+1	60	--
3f*	26	3.33	+1	60	--

* Fluid contained water by infrared analysis - No change by gas chromatograph

AD-A111 319 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 1/2
FIREPROOF BRAKE HYDRAULIC SYSTEM.(U)
SEP 81 S M WARREN, J R KILNER F33615-80-C-2026

UNCLASSIFIED AFWAL-TR-81-2080 NL

5 OF 5

END
DATE
FILMED
03-82
DTIC

5 OF 5

AD A
111319

1.0 2.8 2.5
2.2
2.0
1.1 1.8
1.25 1.4 1.6

APPENDIX J

KC-135 DATA

Data used in the KC-135 real time simulation is compiled below.

J.1 BASIC AIRCRAFT CONFIGURATION DATA

1. WING

   a) Area, theoretical — 2433.0 ft^2
   b) Means aerodynamic chord — 241.8 inches
   c) Angle of incidence — +2.0 degrees

2. HORIZONTAL TAIL

   a) Stabilator deflection (with respect to wing chord plane)
      Minimum - -16 degrees
      Maximum - -1.5 degrees
   b) Elevator deflection
      Minimum - -25 degrees
      Maximum - +15 degrees

3. SPOILERS/SPEED BRAKES

   a) Deflection — 0-60 degrees

4. FLAPS

   a) Landing flaps — 50 degrees

5. POWER PLANT Four, J-57-39N or J-57-43WB

   a) Thrust axis (with respect to waterline) 0.0 degrees (assumed)
   b) Center of thrust, fuselage station — at C.G. (assumed)
      waterline — 193.6 inches
   c) Idle thrust (see Figure J-1)

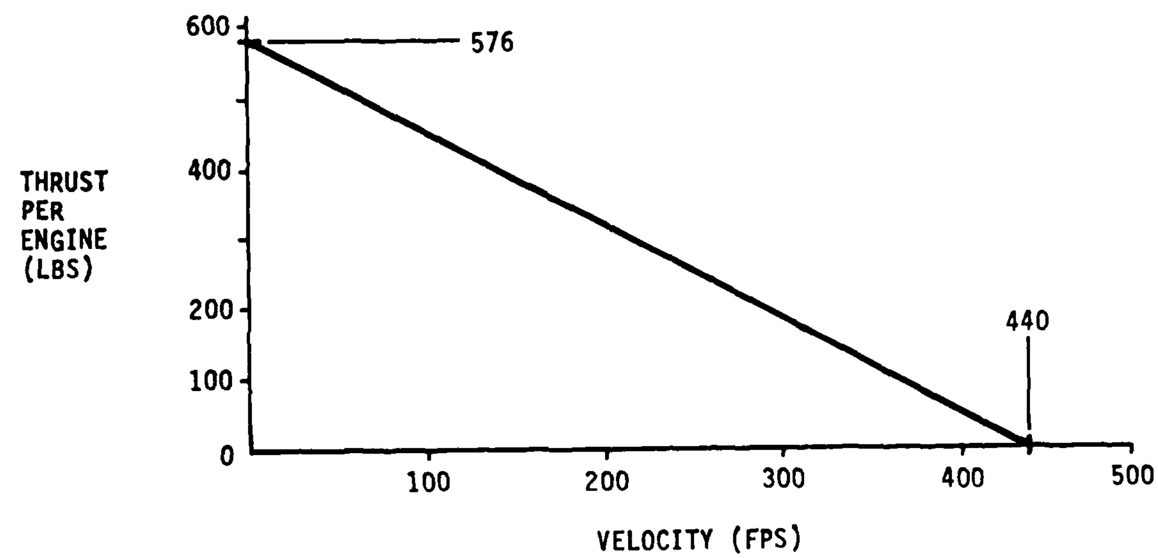


Figure J-1 Engine Idle Thrust

6. NOSE LANDING GEAR CONFIGURATION

a) Attachment point, oleo centerline

	fuselage station	339.0 inches
	waterline	147.16 inches
b) Extended strut length		55.7 inches
c) Gear inclination		0 degrees

7. MAIN LANDING GEAR CONFIGURATION

a) Attachment point, oleo centerline

	fuselage station	887.0 inches
	waterline	177.6 inches
b) Extended strut length		89.19 inches
c) Gear inclination		0 degrees

8. NOSE GEAR TIRE

a) Size	38x11 TYPE VII
b) Inflation pressure	130 psi
c) Load-deflection curve (see Figure J-2)	
d) Undeflected tire radius	19.0 inches

9. MAIN GEAR TIRE

a) Size	49x17 TYPE VII
b) Inflation pressure	150 psi
c) Load - Deflection curve (see Figure J-2)	
d) Undeflected tire radius	24.15 inches
e) Tire Torque Radius	22.02 inches (assumed)
f) Tire Rolling Radius	23.44 (assumed)

10. NOSE LANDING GEAR CHARACTERISTICS

a) Oleo load-stroke curve (see Figure J-3)

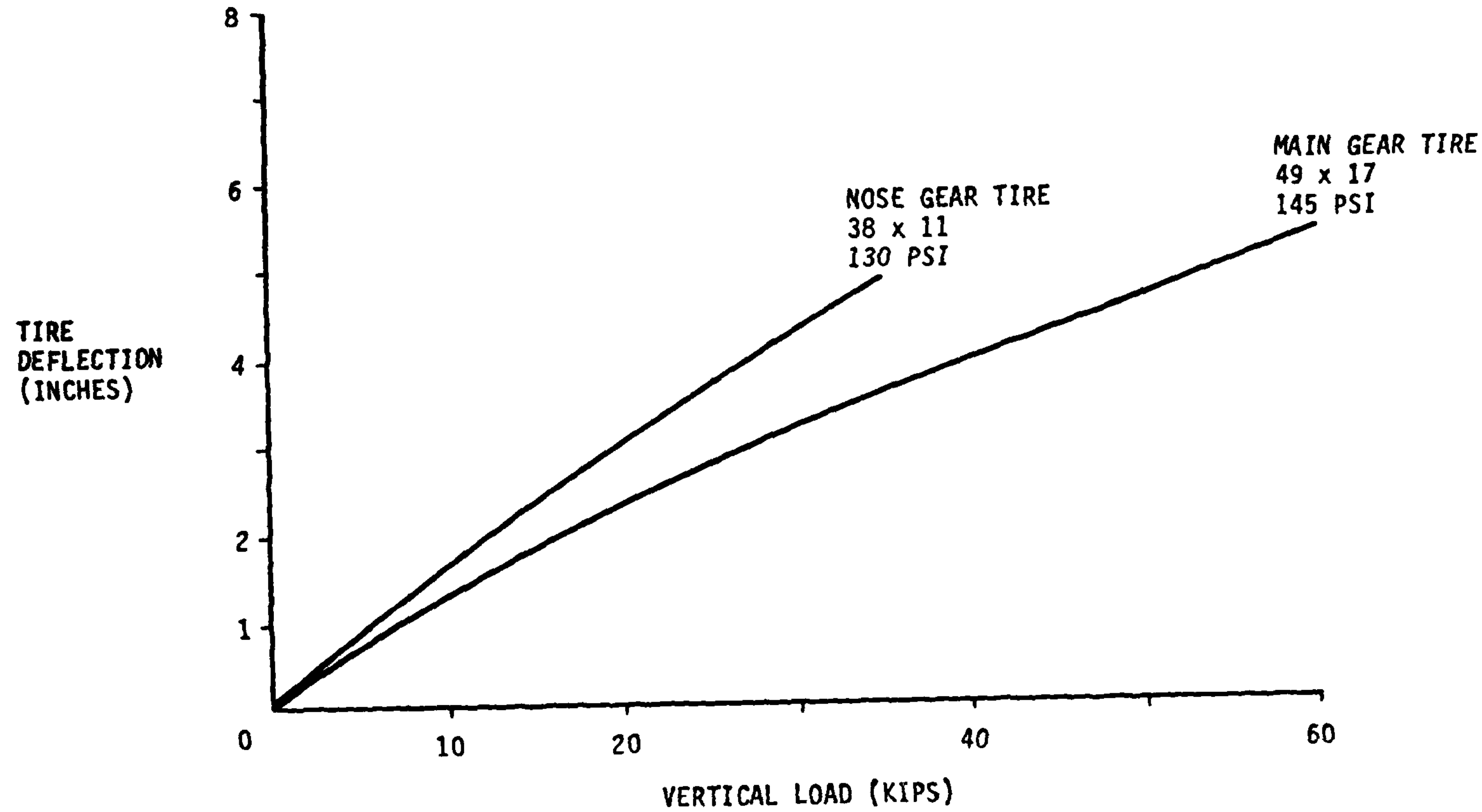


Figure J-2 Line Load-Deflection Curves

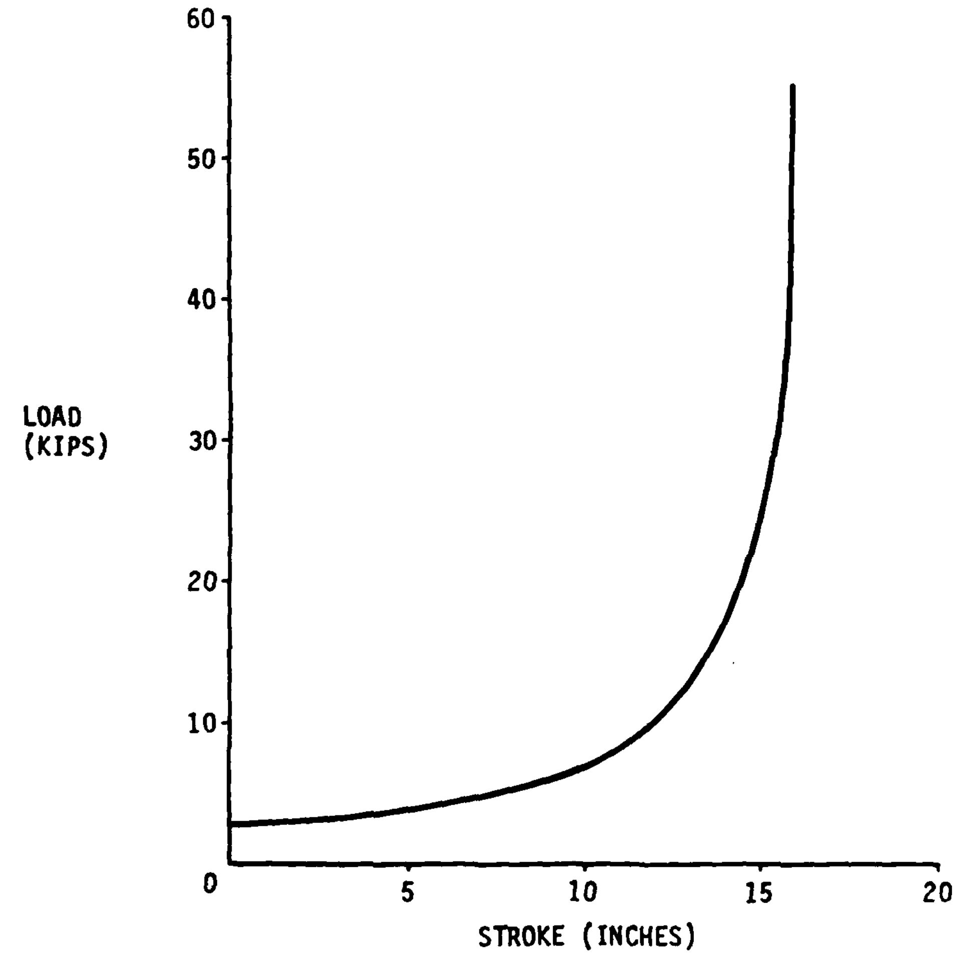


Figure J-3 Nose Gear Oleo Load-Stroke Curve

11. MAIN LANDING GEAR CHARACTERISTICS

a) Unsprung mass, weight 2842 pounds

b) Oleo load-strike curve (see Figure J-4)

c) Fore-aft natural frequency 17.75 Hertz

d) Fore-aft damping ratio 0.1

12. WEIGHT. CG, AND MOMENT OF INERTIAL DATA

J.2 KC-135 AERODYNAMICS

Statistical curve fitting techniques were used to develop relationships (equations) between dependent and independent aerodynamic variables. The equations used in the KC-135 longitudinal aerodynamic simulation are given in in the digital computer aerodynamic subroutine listing, Table G-2.

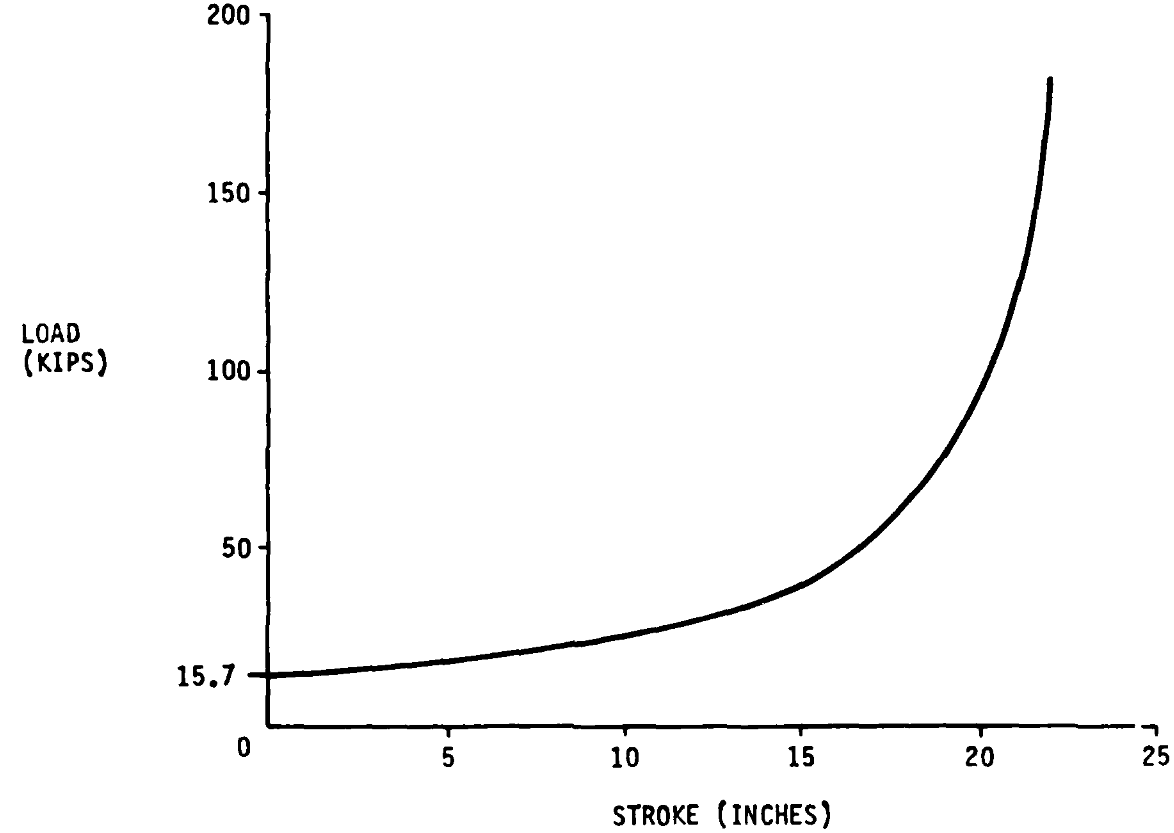


Figure J-4 Main Gear Oleo Load-Stroke Curve

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LIST OF ABBREVIATIONS AND SYMBOLS

CRT	Cathode ray tube
CTFE	Chlorotrifluoroethylene hydraulic fluid
db	Decibels
DOF	Degree(s) of freedom
EMT	Elapse maintenance time
F	Fahrenheit
FPS	Feet per second
FT	Feet
GFP	Government furnished property
HSFR	Hydraulic System Frequency Response
HYBCOL	Hybrid Brake Control Laboratory
HZ	Hertz
IN	Inch(es)
LB	Pounds
MILSTRIP	Government requisition system
MMH	Maintenance man hours
PBM	Pressure bias modulation unit
PNF	Phosphonitrilic fluoroelastomer
PSI	Pounds per square inch
SEC	Second
TEMP	Temperature

*U.S.Government Printing Office: 1981 — 559-003/2011

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